System Evaluation Methods

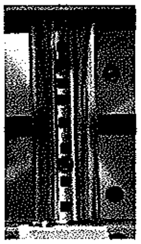


Figure 3: Vibration Distribution measurement

Several preliminary experiments have been performed to examine the characteristics of the system and its performance. A laser vibrometer (Polytec PI, H-300) was used to examine the distribution of sinusoidal acceleration on the handle vibrating at 2g in three directions, as shown in Figure 3a. The system was used to simulate 3-D sinusoidal vibration, a broadband random vibration from 7.5 Hz to 500 Hz, and a cutting saw vibration spectrum.

Evaluation Results

Figure 4 shows the distribution of the vibration on the handle. The maximum difference of the distribution along the handle longitudinal direction in the frequency range (<500 Hz) of concern was less than 9%.

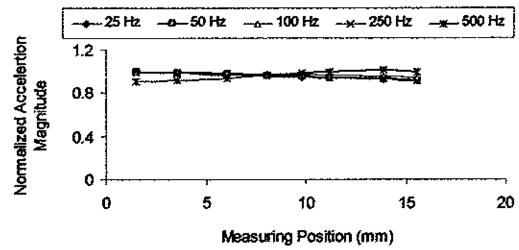
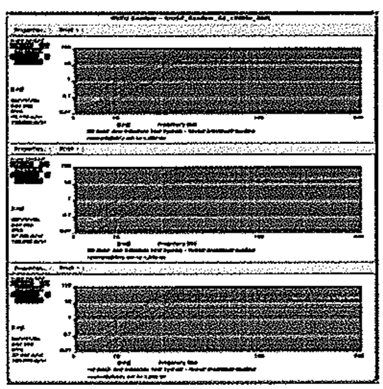
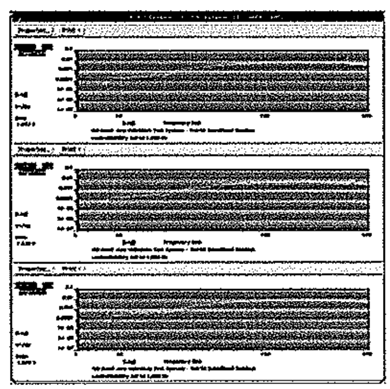


Figure 4: Vibration distribution on the handle

As an example, Figure 5 (a) and (b) display the Control and Drive plots demonstrating full performance. Overall noise levels due to the 10 g's RMS vibration on each axis exceeded 96 dBA in a 52 dBA ambient environment absent the vibration.



(a) Control signal



(b) Drive signal

Figure 5: System performance

Conclusion

These preliminary results suggest that it is acceptable to use the 3-D test system to simulate the sinusoidal, broadband random, and time-history vibrations.

MULTI-AXIS HAND-ARM VIBRATION TESTING & SIMULATION AT THE NATIONAL INSTITUTE OF INDUSTRIAL HEALTH, KAWASAKI, JAPAN

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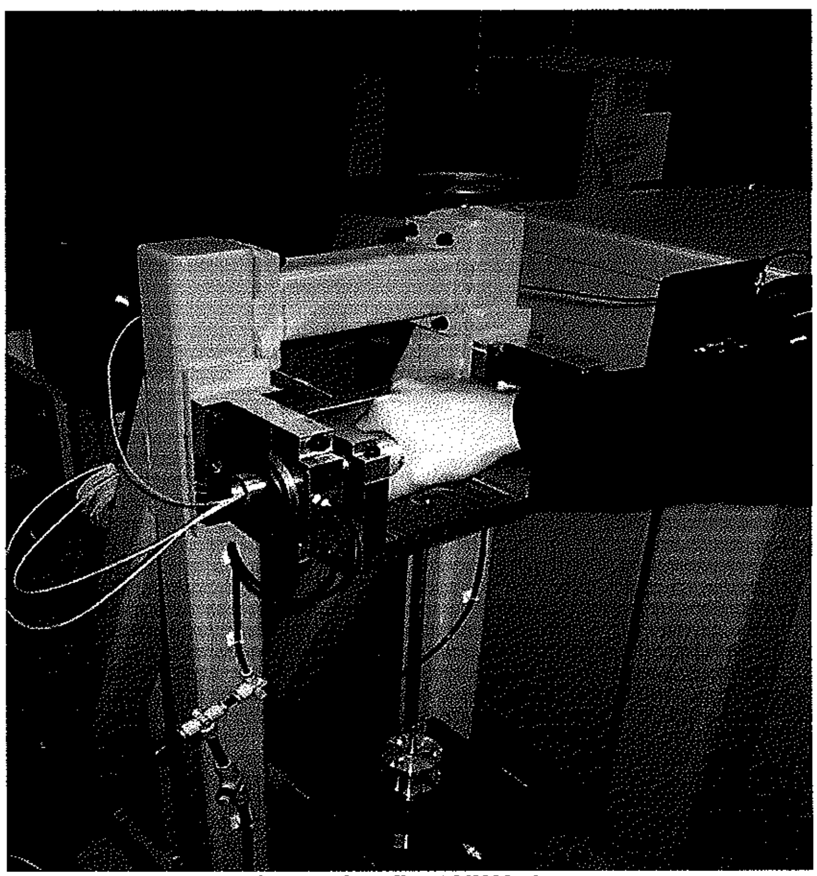
Tony Keller, Spectral Dynamics, Inc., San Marcos, California, U.S.A.

Introduction

Hand-Arm Vibration Syndrome (HAVS) was identified as early as 1918 in Bedford, Indiana in the U.S. Since then much research work has been done around the world in the areas of medical, epidemiological, engineering and legal aspects of HAVS. In Japan, much of the pioneering work in this field has been performed by Dr. Setsuo Maeda and his staff at the National Institute of Industrial Health (NIIH) in Kawasaki. Most recently, reports of work done by this group and by Dr. Ren Dong[1] of NIOSH in the U.S., as well as many other suppliers and Japanese practitioners were presented at the 13th Japan Group Meeting on Human Response to Vibration held in Osaka[2] during August 3-5, 2005.



Patient grasping test handle at NIIH, Japan

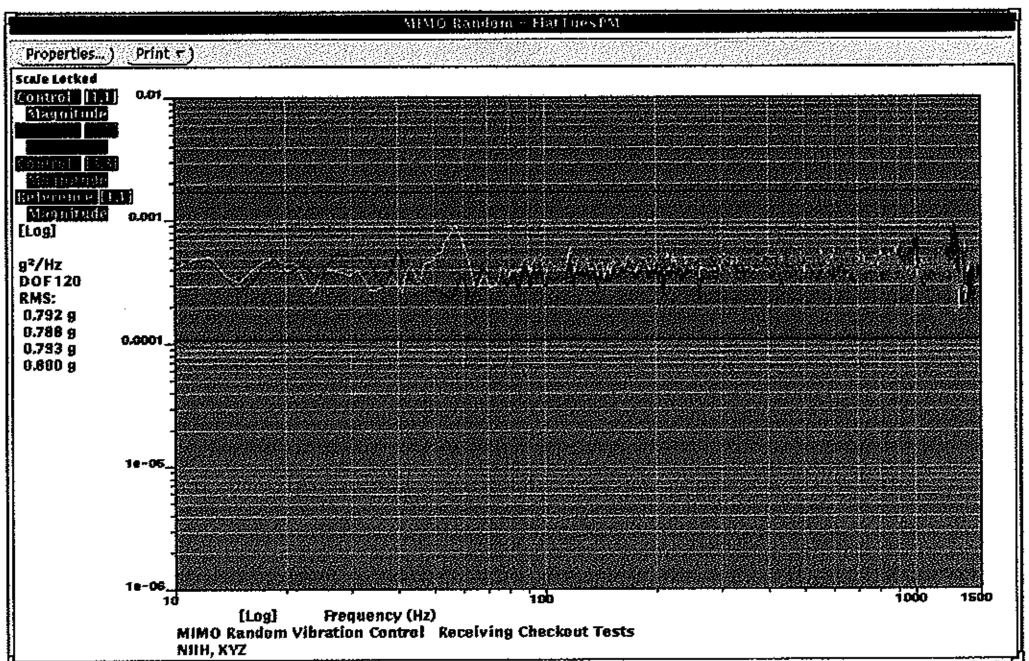
The laboratory at NIIH has been at the forefront of much of the testing technology and instrumentation verification involved in the latest HAVS research which is taking place. An example of this is the recently installed 3-axis vibration simulator in the NIIH laboratory. What follows is a brief description of this system and some results obtained to date.

Methods

Specific methods of measurement and analysis were under development as this abstract was prepared. The presentation may include actual patient response data if it is available at that time.

Results

Results of simultaneous X, Y, Z controlled excitation, like this example, are given.



X, Y, Z Responses controlled from 10 to 1,500 Hz

Discussion

Development is continuing on a modified special handle with embedded Force and Acceleration transducers to understand fully the patient HAVS responses.

References

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2. Keller, T (2005). Some aspects of multi-shaker/multi-axis MIMO. 13th Japan Group Meeting on Human Response to Vibration; Osaka, Japan, 3-5 August, 2005 (JGHRV)

A PILOT STUDY OF THE TRANSMISSIBILITY OF THE RAT TAIL COMPARED TO THAT OF THE HUMAN FINGER

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Introduction

Continual occupational exposure to vibrating hand tools can damage the neural, vascular and other soft tissues of the fingers. Rat tail models have been developed to investigate the biological responses of the tissues to vibration.[1-2] However, the biodynamic response of the tail relative to that of the human fingers has not been characterized. The objective of this pilot study was to compare the transmissibilities of rat tails measured via a scanning laser vibrometer to those of human fingers gripping a handle.

Methods

In Part I of this experiment, four male Sprague Dawley rats (6 weeks old) were exposed to discrete 5g-rms sinusoids of 32, 63, 125, 160, 250, and 500 Hz. The rats were restrained in Broome-style restrainers with their tails constrained without compression to an exposure platform via elastic straps as shown in Figure 1. The platform was attached to a vertically vibrating shaker. The vibration was measured for the array of points shown in Figure 1 using a scanning laser vibrometer (Polytec) and the transmissibility calculated for each point on the tail relative to the reference points on the platform.

In Part II, three male human subjects were exposed at the frequencies specified in Part I - with the addition of 1000 Hz - at a magnitude at the ANSI <0.5-hr limit up to 63 Hz, after which the acceleration was held constant at 5g-rms. The subjects gripped an instrumented handle at 20 N as shown in Figure 2. The transmissibility was calculated relative to the reference points on the handle.

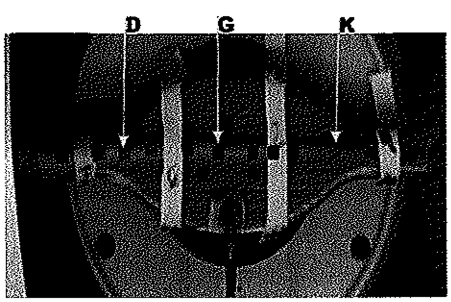

Fig. 1. Index points for tail. D is closest to the rat body.

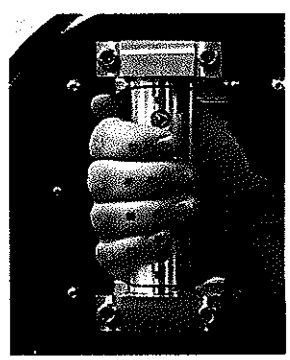

Fig. 2. Experimental set-up for Part II.

Results

Figure 3 shows the transmissibility calculated at the nails of the index, middle and ring fingers of the human subjects. Figure 4 shows a comparison of the transmissibilities of the three most active points on the rat tail with the mean response of all of the tips of the human fingers.

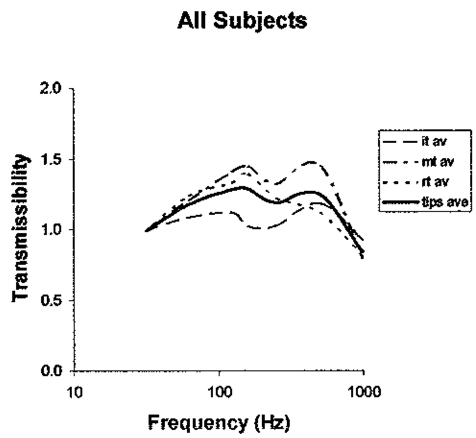

Fig. 3. Transmissibility at middle (mt), and index (it) finger nails.

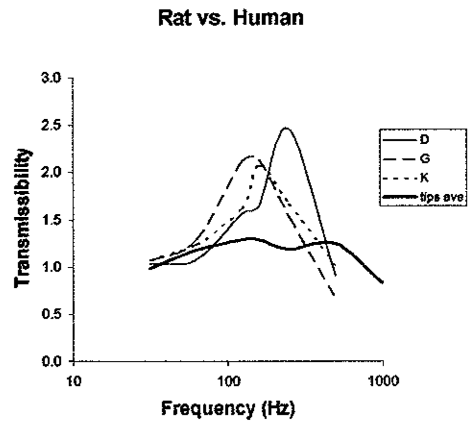

Fig. 4. Comparison of frequency ring (rt) responses of the tail model and the average for the finger nails.

Discussion

As shown in Figure 3, the finger nails tend to show similar frequency responses with comparable first resonances around 125-160 Hz and a second peak at 500 Hz, albeit with varying levels of amplification. The fingers are larger with more mass and damping, while the tail is also stiffer. The rat has considerably higher amplification at all of the most active points. Therefore the rat tail may offer an accelerated model for the investigation of the physiological response to vibration while having similar resonant frequencies to the finger tip.

References

1. Curry et al. (2002). Vibration injury damages arterial endothelial cells. Muscle & Nerve 25, 527-34.
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Podium Presentations

Session VIII: Vibration Reduction and Machine Testing

Chairs: Jack Wasserman and Alan Mayton

SEAT CUSHION AND POSTURE EFFECTS IN MILITARY PROPELLER AIRCRAFT VIBRATION ENVIRONMENTS

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Introduction

Annoyance, fatigue, and musculoskeletal pain have been reported during prolonged exposures to propulsion-generated vibration in military propeller aircraft[1]. The objective of this study was to determine the vibration mitigation properties of selected seat cushions and the effects of occupant seating posture during exposure to higher frequency multi-axis vibration associated with military propeller aircraft.

Methods

A Navy E-2C Hawkeye crew seat was mounted onto the Six Degree-of-Freedom Motion Simulator (SIXMODE). Six seat pan cushion configurations were tested during exposure to an E-2C vibration signal collected in the field[1]. Seat pan cushions 1 – 5 were used with the original E-2C seat back cushion. Cushion configuration 6 included seat pan cushion 5 with a prototype seat back cushion. Triaxial accelerometer pads were mounted onto the seat pan and seat back cushions to measure the vibration entering the human. Data were collected for seven subjects seated upright with their backs in contact with the seat (back-on) and not in contact with the seat (back-off). Spectral analysis techniques were used to analyze data at the two dominant frequencies associated with the propulsion system (propeller rotation frequency (PRF) ~18.5 Hz, and blade passage frequency (BPF) ~73.5 Hz). Overall accelerations were also calculated between 1 and 80 Hz. Vibration Total Values (VTVs) were calculated using the weighted seat pan and seat back (back-on only) accelerations and compared to the comfort reactions given in ISO 2631-1: 1997[2].

Results

In general, the highest accelerations observed at the seat pan occurred in the fore-and-aft (X) direction at both the PRF and the BPF for all cushions and both postures. The most pronounced effect was at the BPF in the X direction, where all configurations showed significantly lower seat pan accelerations than configuration 1 (original E-2C cushion) with the back-on posture. Configuration 5 was the exception with the back-off posture (Fig. 1A, Repeated Measures ANOVA, $P<0.05$). The most pronounced effect of posture occurred at the PRF in the X direction, where all cushion configurations showed significantly lower seat pan accelerations with the back-off posture (Fig. 1B).

All configurations except configuration 2 showed similar VTVs as compared to Configuration 1 (Fig. 2, $P<0.05$). Configuration 2 tended to show the lowest weighted acceleration levels. The overall VTVs (back-on only, Fig. 2B) showed significantly higher accelerations as compared to both the back-on and back-off seat pan point VTVs (Figs. 2A &

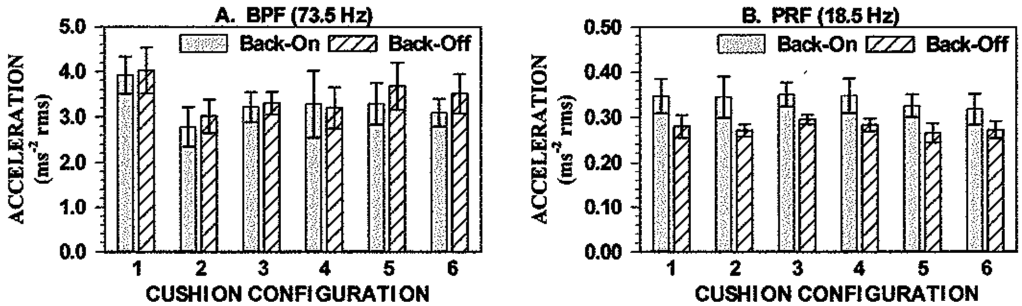

Figure 1 Mean Seat Pan X Accelerations +/- One Standard Deviation at the A. BPF and B. PRF

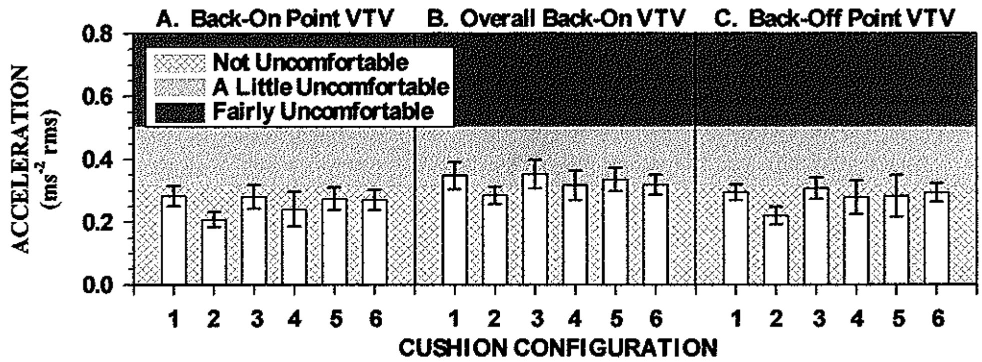

Figure 2 Mean VTVs +/- One Standard Deviation

2C) (Paired t-test, $P<0.05$). Configurations 3, 4, & 6 showed significantly higher back-off point VTVs (Fig. 2C) as compared to the back-on point VTVs (Fig. 2A). Figures 2B & 2C suggest that, in several instances, vibration would be considered at least "a little uncomfortable."

Discussion

The psychophysical effects reflected in the VTVs indicated that the occupants may only perceive a reduction in the vibration with Configuration 2, regardless of the unweighted results. It is noted that the ISO comfort reactions are based on public transport and may not reflect aircrew comfort perception during prolonged exposures. Posture, relative to sitting in contact with the seat back (back-on), does appear to have a significant effect on the vibration. Although not shown, the highest unweighted seat back vibration occurred in the vertical direction, while the highest weighted seat back vibration was estimated to be in the X direction (back-on). These results render it difficult to determine an appropriate strategy for reducing discomfort by mitigating higher frequency vibration through seat cushion design alone. Newer seat designs (active or semi-active vibration isolation systems) may improve seating comfort during prolonged vibration exposures.

References

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(Approved for public release; distribution unlimited. AFRL/WS-06-0257 31 Jan 2006)

COMPARISON OF ANTI-VIBRATION INTERVENTIONS FOR USE WITH FASTENING TOOLS IN METAL

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Introduction

Tool manufacturers continue to incorporate new designs to the internal mechanism of tools in order to decrease the vibration that is delivered to the hand during operation. Modification of some tools to minimize tool vibration is not easily resolved through internal tool design. For this reason, vibration damping materials applied between the tool and the hand are a simple alternative. The damping materials may be applied to the area of the tool directly contacted by the operator or in a glove containing a vibration absorbing pad. These interventions are developed specifically to damp vibration but are not necessarily produced and tested under the same work conditions that a company may expose their workers. Therefore, it is important to test the value of the proposed interventions for the specific applications. This study evaluates the effectiveness of anti-vibration interventions currently in use at a local manufacturing company.

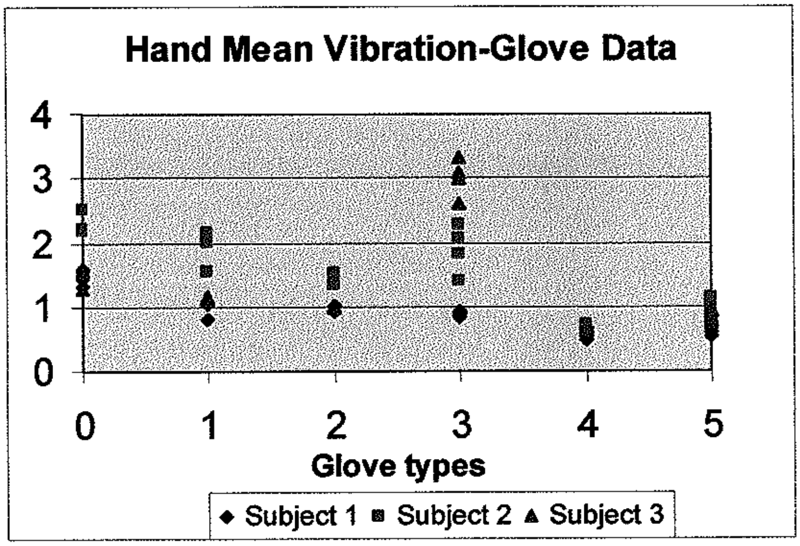
Methods

The design of this study evaluates the vibration energy produced at the tool handle and from the back of the operator's hand. Each operator performed a series of fastener installations in metal using several interventions and one series with no intervention as a control. Four of the interventions were gloves containing anti-vibration material and the fifth intervention was an anti-vibration material wrapped around the tool handle. The protocol for wrapping the tool handle was developed and is part of the equipment procedure at this manufacturing company. Test conditions mimicked production work conditions including similar materials, fasteners and technique for installation. Vibration values were collected using 3 tri-axial accelerometers with one firmly glued to the tool handle close the hand grip as recommended by ISO 5349. A second accelerometer was placed on the top of the pistol shaped tool. The third accelerometer was attached to the back of the hand close to the third knuckle using double sided tape.

Results

Preliminary results for 3 volunteers show a difference between the vibration values of the control condition (mean hand vibration on bare hand = 1.77 Gs) compared to all of the interventions ($p=.0001$ using Mixed Procedure, Tests of Fixed Effects).

The graph below shows individual trials for each subject for each condition. Glove 3 with the air bladder insert shows large variability between subjects (Range = 0.84-3.33 Gs). The other interventions show much less variability both within subjects and between subjects indicating consistent response with use of the intervention.



Glove Types
0 = bare hand
1 = mesh
2 = wrist support
3 = air bladder
4 = foam
5 = wrapped handle

Discussion

All interventions showed less vibration energy produced at the level of the hand compared to the control condition. Thus providing an interface between the operators hand and vibration source decreased the energy directed to the operator. Three of the gloves produced a beneficial response with minimal variation. Intervention glove 3 containing the air bladder provided less consistent beneficial effects due to the large variability in response. This device requires the operator to manually pump the air bladder to the desired level. The manufacturer recommends pumping the air bladder 50 repetitions prior to the initial use and a few additional pumps each day the glove is used. The amount of air delivered to the glove for this pilot was determined by the personal preference of the subjects and resulted in large variability in vibration output.

Intervention 5 consists of a Viscolas™ material wrapped over a tool handle, and held in place with shrink wrap. The manufacturing company developed this method to provide protection to the workers with a durable wrap that was cosmetically pleasing. The lowered vibration values for the Viscolas™ wrapped handle compared to the control indicates the method of wrapping the handle is protective to the operator.

Since both the gloves and the Viscolas™ wrap on the handle of the tool measured lower vibration values, work conditions and behaviors of the workers should be considered to determine the recommended intervention. Use of gloves to minimize vibration exposures requires the operator's consistent use of the glove during all tool use. Wrapping the handle of a tool to protect a worker from vibration exposures does not depend upon a worker's behavior for effectiveness. Assuming all areas of the tool encountered by the hand are covered with the Viscolas™ material, every time the worker grasps the tool, the hand is protected. Since three of the gloves in the study are fingerless, the anti-vibration material will not protect the exposed skin. Operators cannot manipulate small fasteners with full fingered gloves. Recommendations for anti-vibration materials for use in a work force should consider the work methods and behaviors of the operators. In determining a recommended intervention for a particular manufacturing process, it is important to test the real physical conditions as well as the typical behaviors of the workers.

VIBRATION CONTROL ON HAND-HELD INDUSTRIAL POWER TOOLS

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Product Ergonomics, Atlas Copco Tools & Assembly Systems, Stockholm, Sweden

Introduction

Work with hand-held power tools can be found in most industries all over the world. This type of work exposes the operators to different kind of loads like gripping-forces, feed-forces, exposure to vibration and noise, holding hot or cold surfaces and the exposure to dust. Designing a power tool with good ergonomics is a matter of finding the best compromise. As a simple example, increasing the mass is not acceptable because it will increase the forces needed to handle the tool. At the same time increased mass will in most cases reduce the vibrations.

Vibration disorders related to the use of hand-held power tools has been known and reported since long. It is therefore essential that low vibrating tools are developed and used. The new vibration regulations in Europe, based on the Physical Agents (Vibration) Directive, have put increased focus on the vibration control in industry.

Forces acting on the tool cause vibration

Tools for industrial use must be of very robust design to withstand the very hard use they are exposed to. Industrial tools are therefore normally designed with the main parts made of metal. From a vibration point of view this means that most tools can be regarded as rigid bodies, especially because the dominating frequency normally is equal to the rotational frequency of the tool spindle or the blow frequency for a percussive tool. These frequencies are with few exceptions below 200 Hz. Handles however can not always be regarded as rigidly connected to the tool. There are several examples of weak suspensions designed to reduce vibration transmitted to the hands of the operator. There are also examples of designs where the handles just happened to be non-rigidly connected and in some cases even in resonance within the frequency region of interest. Oscillating forces act on the tool and the result is vibration.

Design principals

In all cases forces are the source of vibration. This leads to the three basic principles to control vibration:

- **Control the magnitude of the vibrating forces.** Examples are the balancing unit on a grinder or the differential piston in a chipping hammer.
- **Make the tool less sensitive to the vibrating forces.** Examples can be when the mass of the guard on a grinder is rigidly connected to the tool to increase the inertia of the tool.
- **Isolate the vibrations in the tool from the grip surfaces.** Examples are vibration-dampening handles on grinders or pavement breakers, the air-spring behind the blow-mechanism in a riveting hammer or the mass spring system in a chipping hammer.

Control the magnitude of the vibrating forces

For rotating machinery the balance of the rotating parts is essential. The inserted tools that will be mounted on the tool spindle often give major contribution to the unbalance of the rotating parts. This is a problem because the tool manufacturer has no control over the inserted tools. The only thing that can be done is to design flanges and guides to fine tolerances as close as possible to the tolerance interval for the inserted tool.

Limiting the power of the tool will in most cases also reduce vibration but that is not a possible route because lower power leads to increased usage-time to get the job done and that would negatively affect the daily exposure.

Make the tool less sensitive to the vibrating forces

A tool will be less sensitive to oscillating forces when mass and or inertia is increased. To increase mass can be questioned from an ergonomic perspective. In some cases when a small increase in mass give a big increase in inertia it might still be a good solution. The tool can be regarded as a rigid body suspended in weak springs. Therefore it will move around its centre of rotation. The perpendicular distance between the forces acting and the centre of rotation will determine how the pattern of movement will be. By altering this distance the movement of the tool can be controlled.

Isolate the vibration in the tool body from the grip surfaces

To isolate the handles from the vibration in the tool body is the most common thing to do. Modern chain saws and breakers are examples where this principal have been successfully applied. The mass spring system must be designed to have the excitation frequency from the vibration well above the systems resonance. This requires a certain mass in the handles or the spring need to be very soft. A correlated problem is when mass is moved from the body of the tool to the handles. The reduced mass will make the tool-body more sensitive to the vibrating forces and the vibration amplitude in the body will increase.

Summary

An industrial powertool can in most cases be regarded as a rigid body. The handles are not always part of this rigid body.

- Forces acting on this rigid body are the source of vibration. The forces are either forces from the process or process independent e.g. unbalances in rotating parts.
- There are three basic principals for vibration control. Control the magnitude of the vibrating forces. Make the tool less sensitive to the forces. Isolate the vibration in the tool body from the grip surfaces.
- All three principals are used in vibration control on power tools either one by one or combined on the same tool.

VIBRATION EMISSION MEASUREMENT METHODS FOR GRINDERS

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Atlas Copco Tools & Assembly Systems, Stockholm, Sweden

Introduction

ISO8662-4, "Hand-held portable power tools - Measurement of vibrations at the handle - Part 4: Grinders" is under revision. The new revision shall harmonize ISO 20643 "Mechanical vibration - Hand-held and hand-guided machinery - Principles for evaluation of vibration emission" which, among others, requires measurements in three directions and declared values related to the upper quartile of real-use vibration.
To get the most suitable test method, a round robin test was made for evaluation of the two test methods proposed by the ad-hoc group working with this standard revision.

Methods

Seven laboratories measured the vibration from four grinders of different sizes, with and without autobalancing units. The laboratories come from universities, health & safety laboratories and grinder manufacturers.

Two measurement methods are evaluated with respect to repeatability and reproducibility:

1. Grinding on a well-defined mild steel bar with depressed center wheels according to detailed test instructions. The test sequence starts and ends with 10 seconds of running the grinder in the air, when measuring the unbalance contribution to the vibration coming from the unbalance of the grinding wheel. Between these runs the average vibration during 60 seconds of grinding is measured. Three operators do five grinding tests per grinder.
2. Measurements using an aluminum unbalance disc similar to the one defined in ISO8662-4. Each operator runs the grinder four times, between each run the unbalance is moved 180 degrees to avoid variations caused by the play between the test wheel and the spindle. The averaging time is 10 seconds. Each grinder is tested by three operators.

Repeatability is the spread within a lab between operators and over short time period for one machine and reproducibility is the spread between laboratories and over longer time periods for one machine. Instrumentation and transducer location are chosen according to ISO8662-4 and circulated test instructions.

Results

Both the repeatability and reproducibility is poor for the real grinding test, see figure 1. The coefficient of variation for repeatability is approximately 40% higher for the grinding test and the coefficient of reproducibility is 60% higher for the grinding test than for the unbalance disc test.

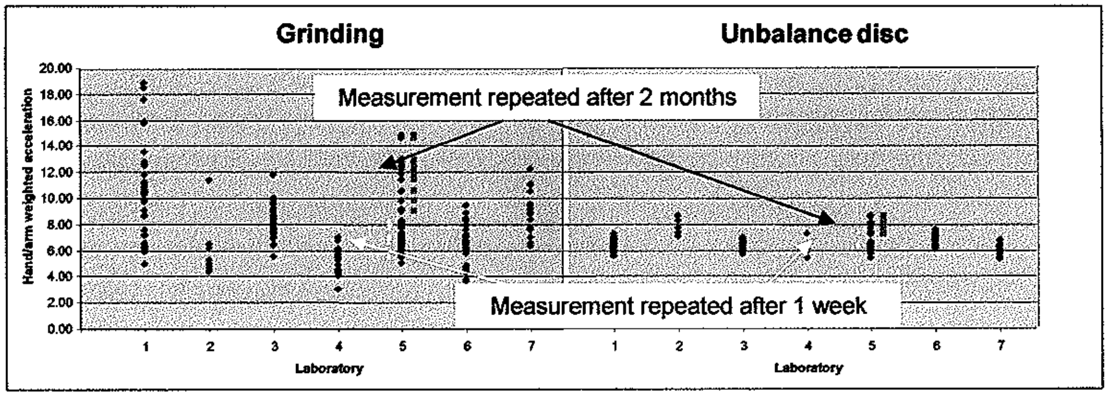

Figure 1. Example of result from grinding test and unbalance disc test. The grinding test shows a larger spread between test runs, operators, and laboratories and over time.

The unbalance disc test gives vibration values corresponding to the upper quartile of the real grinding test for grinders without autobalancing units. This is one requirement in the revised vibration measurement standard. Grinders with autobalancing unit gives lower values for the unbalance disc test, therefore they require additional grinding tests to fulfill this requirement.

Discussion

Unbalance disc test is proven to be the most accurate method for measuring vibrations from grinders, with one exception; grinders with autobalancing units. The result from this study also shows that it is extremely time consuming to get reliable field vibration measurements on grinders. The result is varying depending on many factors that are difficult to control; feed force, grinding wheel quality, work piece etc. The unbalance disc test gives values with good repeatability and reproducibility which well correspond to the upper quartile of the vast amount of grinding measurements made in this study. Thus, it is recommended to use the declared value according to ISO 8662-4 when assessing the vibration emission from grinders instead of doing field measurements. When using emission values from manufacturers, it is important to verify that the value is measured according to appropriate ISO-standard.

References

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COMPUTATIONAL SIMULATION OF A PNEUMATIC CHIPPING HAMMER

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Introduction

Occupational exposure to hand transmitted vibration (HTV) arises from the hand held powered tools extensively used in the mining and construction industry such as rock drills, chipping hammers, chain saws etc. Regular exposure to HTV is the major cause of a range of permanent injuries to human hands and arms which are commonly referred to as hand-arm vibration syndrome (HAVS). In addition to this, the percussive tools generate overall sound power levels in excess of 110dBA in most cases. Such a high sound power level greatly exceeds the maximum permissible exposure limit (PEL) of organizations such as National Institute for Occupational Safety and Health (NIOSH) and the Occupational Safety and Health Administration (OSHA). Long term occupational exposure to this noise has been diagnosed as the main reason for permanent hearing loss in the operators. It is therefore important to develop an understanding of the mechanisms which lead to these high vibration and sound levels and in order to do this a detailed computational model of a pneumatic chipping hammer has been made.

This paper presents a nonlinear computational model of a pneumatic chipping hammer. In order to better understand the dynamics of the chipping hammer, the hammer was subdivided into components that are shown in figure 1 (a) (based on a chipping hammer manufactured by Atlas-Copco). The hammer mainly consisted of a center body, a moving piston and a chisel. Compressed air is used to drive the piston inside of a cylinder and on the downward stroke this piston impacts the chisel to create the hammer effect. The machine has one pneumatic valve and this valve regulates the air supply either to the upper chamber or to the lower chamber. The valve changes according to the relative pressures in the two chambers and the supply pressure. There are also twelve different exhaust ports at two positions along the cylinder labeled upper ports and lower ports. As the piston moves the ports can be closed or open (allowing exhaust).

Fundamentally, the computational model was made up of two different sub-models, a fluid model and a structural dynamic model as shown in Figure 3 (a) and (b) respectively. The first sub-model takes into consideration the fluid dynamics of the machine since the hammer is driven by compressed air. Equations for the mass flow rate though bleed orifices (assuming an isentropic process) [1] is used to determine the mass flow into and out of the upper and lower chambers. From this the pressures in the two chambers and consequently the forcing on the piston can be calculated. The second sub-model deals with modeling the structural

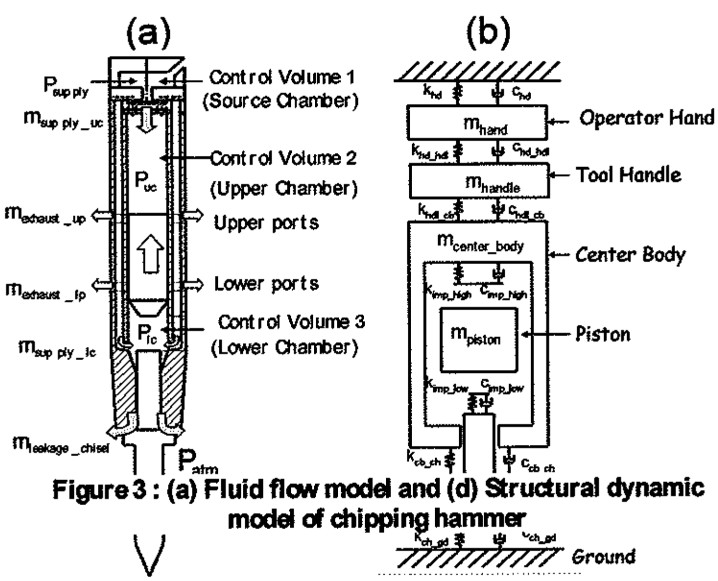

Figure 3 : (a) Fluid flow model and (d) Structural dynamic model of chipping hammer

components of the chipping hammer. The structural model consists of various lumped masses[2], each representing a specific component of the chipping hammer as well as the ground and operator's hand. The impact dynamics were also incorporated by connecting the piston and the chisel with a non-linear spring. The fluid flow and structural models were then coupled together using a time domain, state space formulation to compute the displacements of each component, the pressures in the chambers, the impact forces and the jet velocities from the exhaust ports. The computational model was then validated using experimental obtained vibration levels and exhaust velocities.

Results

Figure 4 (a) and (b) show the experimental and computational exhaust velocities from the upper and lower exhaust ports respectively. There is a very good match between the exhaust jet velocities measured during lab tests and the exhaust jet velocities calculated from the computational model. Also the tool impact frequency measured from lab tests is approximately 27 Hz which is very close to the tool impact frequency calculated from the computational model (32Hz). Keeping in mind the nonlinear nature of the fluid flow model, these can be considered as good results. However, further refinement of the fluid flow model will be continued in the near future. The structural dynamic response of the computational model will be discussed at the time of presentation.

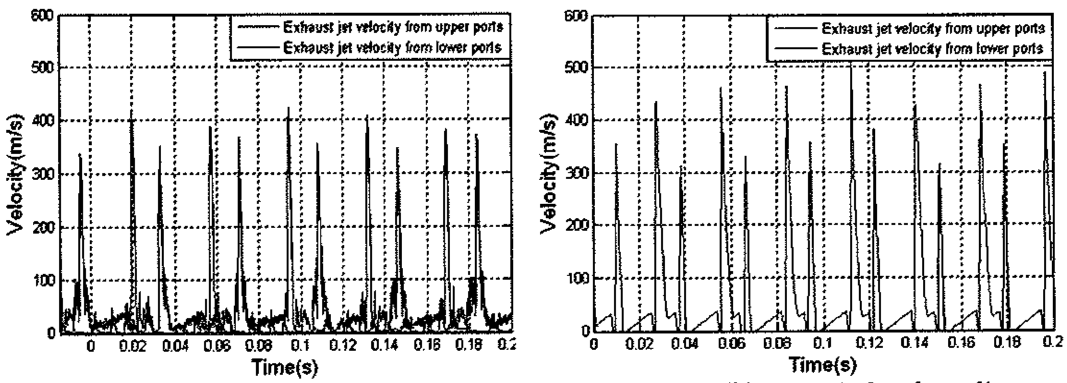

Figure 4 : Exhaust jet velocities (a) experimental results, (b) computational results

This model provides a unique opportunity to evaluate different vibration and noise control techniques and consequently to help determine the best possible control method. The model would avoid the need for extensive laboratory testing which is time consuming as well as expensive.

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DESIGN OF A TEST BENCH TO EVALUATE THE VIBRATION EMISSION VALUES OF JACKLEG ROCK DRILLS

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Introduction

Jackleg rock drills are widely used in the mining industry and are known to generate high levels of hand-arm vibration which contribute to the development of the hand-arm vibration syndrome for exposed miners.[1-3] To reduce the vibration levels, a prototype of an antivibration handle was developed as part of a previous study.[4] To provide some bench marking for this handle prototype and to follow the evolution of its performance over time, a test bench was developed to characterize the vibration emission values of jackleg drills under controlled operating conditions. As the current ISO 8662 series of standards could not apply directly to this type of tool, there was a need to design and validate a test bench to evaluate the vibration emission values of jackleg drills, while taking into account the conditions specific to the operation of this type of tool.

Methods

A test bench including an energy absorber, was developed for testing jackleg drills based on the ISO 8662-3 standard [5]. The energy absorber was bolted to a 3300 kg concrete block to ensure tool stability. A pictorial view of the device is given in Figure 1. For validation purposes, acceleration measurements at the handle of a conventional jackleg drill were taken simultaneously along the three axes (x_h, y_h and z_h) in an underground rock drilling operation as well as on the test bench. The handle accelerations were measured for three different jackleg angles (13°, 28° and 43°) determined with respect to the floor. Moreover, each measurement was repeated at least three times to asses the data repeatability.

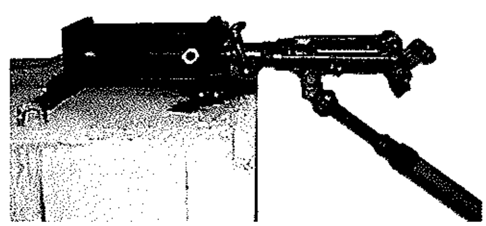

Figure 5. Jackleg drill (right) with the energy absorber (left)

Results

As a preliminary validation of the test bench, Figure 2 provides a comparison of the frequency weighted rms acceleration spectrum measured along the z_h-axis, for both underground drilling and operation on the test bench (28° jackleg angle in both cases). It is shown that the vibration measured on the handle of a jackleg drill operating on the test bench is representative of that recorded during typical rock drilling operations, despite the fact that some harmonics of the percussion frequencies are generated with a higher amplitude on the test bench. Table 1 provides

a comparison of the overall frequency-weighted rms accelerations measured for all three jackleg angles. It is shown that the test bench provides comparable values of overall acceleration for all three axes, with much lower variation coefficients (COV) on the test bench, suggesting a higher measurement repeatability. In addition, it was verified that the measurements obtained on the test bench were reproducible, by ensuring that similar frequency-weighted rms accelerations could be obtained after completely reinstalling the jackleg drill on the test bench.

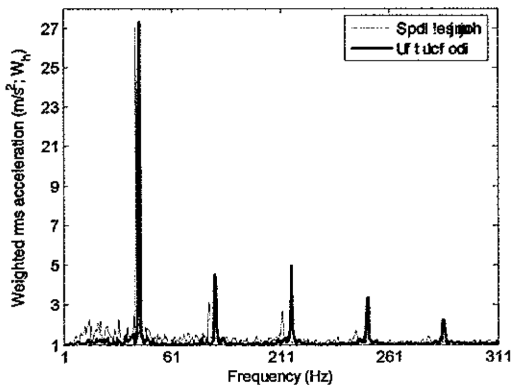

Figure 2. Comparison of vibration frequency spectrum measured on the test bench and while drilling (z_h percussion axis)

Table 1. Comparison of frequency weighted rms accelerations measured for three different jackleg angles on the test bench and while drilling.
m/s^2 (w_h)
COV (%)

		x axis	y axis	z axis	Total
13 deg	Bench	**10.66** 0.84	**5.33** 5.33	**20.72** 1.72	**23.90** 1.41
	Drilling	**12.80** 44.19	**6.18** 2.21	**24.30** 3.52	**28.41** 11.69
28 deg	Bench	**9.51** 1.98	**5.15** 2.03	**19.90** 0.79	**22.65** 0.73
	Drilling	**9.64** 8.96	**6.18** 29.70	**22.70** 13.13	**26.44** 13.33
43 deg	Bench	**11.46** 1.32	**5.62** 0.54	**18.74** 0.48	**22.65** 0.60
	Drilling	**8.78** 11.62	**4.92** 7.27	**19.73** 6.73	**22.16** 7.35

Discussion

The validation of a test bench to characterize the vibration emission values of jackleg rock drills has been presented. Preliminary results have shown that the test bench provides a good representation of the vibration measured during rock drilling operations, while providing a better repeatability of the acceleration values. Thus the test bench appears to be applicable to characterize the vibration emission values of jackleg drills.

Acknowledgements

The authors wish to acknowledge the financial support provided by SOREDEM .

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Podium Presentations

Session IX: Epidemiology, Standards Applications, and Prevention II

Chairs: David Wilder and Kristine Krajnak

RISK ASSESSMENT OF HAND-ARM VIBRATION BY ESTIMATE, TAKING THE EXAMPLE OF HAND-GUIDED STONE-WORKING MACHINES

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Introduction

Vibration measurements at the workplace are often complicated and expensive. The assessment of the risk in conformity with EC Directive 2002/44/EC "Vibration" (which lays down the minimum requirements of laws in Europe for occupational safety and health) can therefore be carried out on the basis of an estimate based on information from manufacturers as well as by measurement conforming to ISO 5349.

The characteristic values (emission values) determined by manufacturers in laboratory conditions may deviate from the exposure values measured at source at the workplace. Equally, deviations may arise as a result of the delay in the changeover of test methods from the single axis of measurement to the total vibration value for the three axes of measurement conforming to ISO 20643.

To prevent faulty estimates, the manufacturer's information has to be corrected by a tool-related factor in accordance with CEN/TR 15350. By taking the example of masonry and stone working machines, the empirically determined tool-related correction factor is checked and confirmed.

Methods

Vibration measurements were carried out in accordance with ISO 5349 in practical application conditions on 10 selected typical eccentric and orbital sanders, concrete and disc grinders as well as on wall chasers and stone saws.

Results

The total vibration value obtained for the investigated tools rangedfrom a_{hv} = 3.6 m/s² to a_{hv} = 11.6 m/s². When the values from the practical measurements are compared with the manufacturer's vibration values, the underestimation of the risk occurring in some cases can be largely compensated for by the tool-related factors conforming to CEN/TR 15350 (see Figure 1).

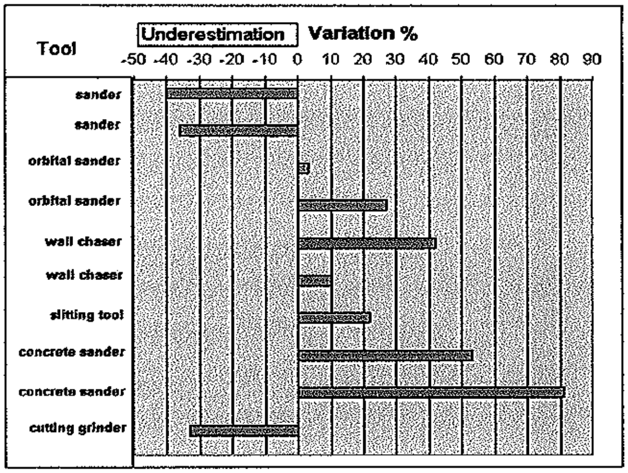

Figure 1.
Variation of the estimated vibration values from the values obtained in practice after correction.

Discussion

The risk assessment can be carried out on the basis of an estimate based on information from manufacturers. The procedure is presented with reference to examples. In three of the ten investigated cases, there was slight underestimation after correction. However, these variations lie within the accuracy range achievable with workplace measurements.

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WHOLE-BODY VIBRATION EXPOSURE AND DRIVER POSTURE EVALUATION DURING THE OPERATION OF LHD VEHICLES IN UNDERGROUND MINING

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Introduction

Load-haul dump vehicles (LHDs) are used to move waste rock and ore in underground mining operations. The LHD is designed for bi-directional operation and the driver sits sideways to the direction of travel. LHD operators have higher reports of low back pain and neck discomfort than other mobile equipment operators who do not sit sideways in the vehicle, but are exposed to whole-body vibration (WBV)[1].

Exposure to WBV is linked with reports of lower-back pain, neck problems and spinal degeneration[2,3]. Static sitting postures, sitting with the neck and back twisted, and sitting with the back in an unsupported posture are also linked with an increased risk of developing back pain[4]. The objective of this study was to determine typical vibration exposure levels and driving postures for LHD operators.

Methods

Whole-body vibration exposure was measured at the seat-pan, in accordance with the ISO 2631-1 standard[5], on seven LHD vehicles with a 10 yard bucket haulage capacity. Vibration data were recorded with a Biometrics™ DataLog II (P3X8) and stored on a 128 Mb Simpletech™ multimedia card. Comparisons were made to the ISO 2631-1 Health Guidance Caution Zone (HGCZ) in order to determine potential injury risk.

Operator posture was monitored with three digital video cameras which were secured inside each operator's cab to the top left corner, top right corner and back right corner. Reflective tape was placed on each driver's shoulders, head, and back in several locations and in several locations on the vehicle seat in order to aid in posture coding. Posture coding was performed with 3DMatch v4.50 multiple video view analysis feature. Vibration measurement and posture recording occurred simultaneously for 60 minutes while the LHD operator performed typical duties.

Results and Discussion

Results indicate LHD operators may be exposed to whole-body vibration levels putting them at risk for injury (Table 1). According to ISO 2631-1 the frequency weighted acceleration values corresponding to the lower and upper limits of the HGCZ (for an 8 hr exposure duration) are 0.45 and 0.90 m/s^2 respectively[5]. Six of the seven vehicles showed exposure levels within the HGCZ defined for 8 hours.

Preliminary video analysis indicated LHD operators were exposed to potentially harmful levels of WBV while adopting asymmetric postures (Table 2). For example, one LHD operator (Figure 1) worked with his neck twisted greater than 40 degrees for 93 % of a 60 minute work cycle. According to the Swedish National Work Injury Criteria, neck rotation should be less than 15 degrees if the motion is required for greater than 80% of the work time[6]. Results of this study highlight the need to further examine the contribution of non-neutral working postures and

WBV exposure in or above the ISO 2631-1 HGCZ given the development of higher than average levels of low back and neck injuries amongst LHD operators.

Table 1: Summary of frequency weighted acceleration (multiplying factor k for health evaluation applied) and the equivalent 8h frequency weighted acceleration (vibration cycle of 7 hours within an 8 hour work day) values for typical underground LHD operation. The axis associated with the dominant value is shown in bold.

Mine & Model	Duration (min.)	aw_x (m/s^2)	aw_y (m/s^2)	aw_z (m/s^2)	a_v (m/s^2)	a_8 (m/s^2)
1 -B	68	0.51	0.45	**0.69**	1.18	0.60
1 -A (1)	70	0.70	0.47	**0.81**	1.44	0.70
1 -A (2)	78	0.68	0.51	**1.01**	1.56	0.83
2 -F	124	**0.67**	0.45	0.63	1.30	0.41
2 -C	117	0.69	0.58	**1.12**	1.68	0.75
3 -C	66	0.65	0.56	**0.78**	1.43	0.69
3 -H	70	**0.61**	0.56	0.56	1.29	0.49

Table 2: Postures adopted along with the percentage of time spent in each posture during a 60 minute monitoring duration, for a typical LHD operator.

Posture Adopted	% time adopted
Neutral neck rotation (< 15 degrees of rotation)	3
Mild neck rotation (15 - 40 degrees of rotation)	4
Severe neck rotation (>40 degrees of rotation)	93
Neutral trunk rotation (< 15 degrees of rotation)	97
Mild trunk rotation (15 – 30 degrees of rotation)	3
Severe trunk rotation (> 30 degrees of rotation)	0
Neutral trunk flexion (< 15 degrees of flexion)	93
Mild trunk flexion (15-30 degrees of flexion)	7
Severe trunk flexion (>30 degrees of flexion)	0
Neutral trunk lateral bend (< 15 degrees of bend)	86
Moderate trunk lateral bend (15–30 degrees of bend)	14
Severe trunk lateral bend (> 30 degrees of bend)	0

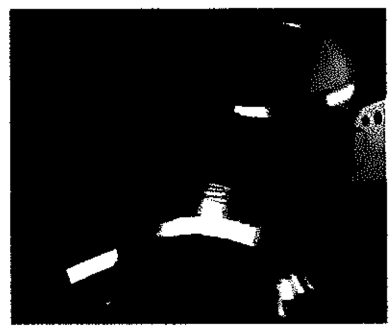

Figure1. Typical posture adopted by LHD drivers.

Acknowledgment

Funding for this research was provided by the Workplace Safety and Insurance Board of Ontario with support from the Mines and Aggregates Safety and Health Association of Ontario and the Ontario Mining Industry.

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MEASUREMENT AND EVALUATION OF VIBRATION EXPOSURE FOR LOCOMOTIVE CREW MEMBERS

Robert Larson, Christine Raasch, Janine Pierce
Exponent, Inc.

Introduction

The vibration and impact environment for crew members on locomotives has been investigated in a series of studies conducted by Exponent Failure Analysis Associates (Exponent) beginning in 1990. Locomotive cab vibration and impact levels were measured on a variety of locomotive models operating over many different track sections across the Union Pacific, Burlington Northern Santa Fe, CSX, Norfolk Southern, and CONRAIL systems. The comfort and health implications of exposure to the measured locomotive vibration levels were evaluated by comparison with the human vibration exposure boundaries given in the International Standards Organization (ISO) standard 2631-1:1997, the British Standard 6841:1987, European Union (EU) Directive 2002/44/EC, measurements made by Exponent on various commercial and recreational vehicles, and vibration exposure measurement data found in the literature.

Methods

Initially, vibration levels experienced by locomotive crews were measured and recorded at incremental speeds covering the range of normal train operation. In 2003, a method of measuring the vibration exposure continuously by means of a digital recorder was developed, allowing the vibration level over the entire run or crew shift to be analyzed. For each seating location measured, acceleration was recorded on the seat surface beneath the ischial tuberosities (pelvis) of the seated crew member and on the cab floor directly under the seat. At each of the locations, triaxial accelerometers were used to measure the vibration along the longitudinal, lateral, and vertical axes. Since the vibration environment varies throughout the route, and locomotive vibration levels have been found to be primarily speed dependent, a speed sensor was used to continuously measure the speed of the train.

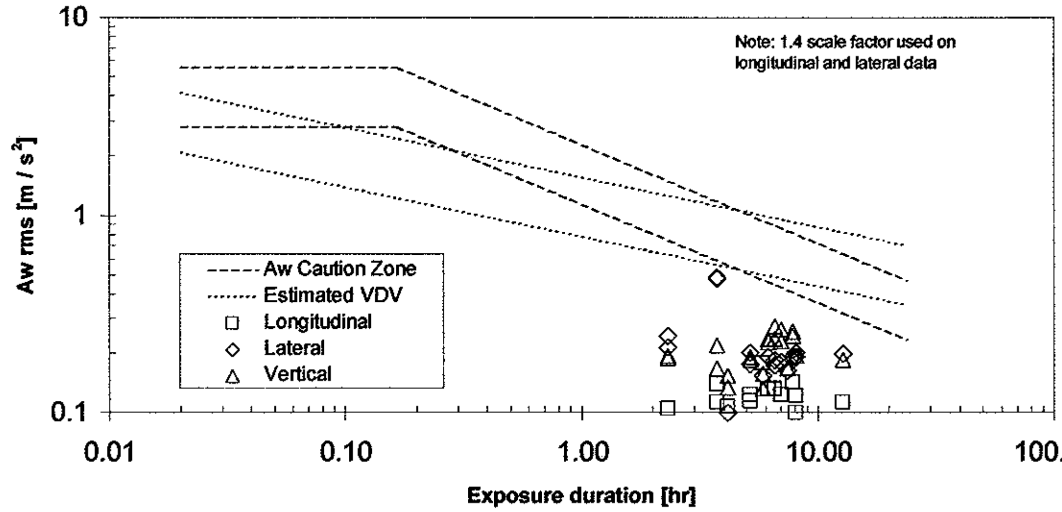
To evaluate the recorded vibrations levels, the data was divided into two-minute segments, which were each processed per the 1997 ISO standard for weighted RMS vibration levels and Vibration Dose Values (VDV). Additionally, PSDs and 1/3 Octave RMS values were calculated to determine the frequency content of the vibration. For each two-minute segment, the average speed of the locomotive was calculated to allow for correlation with the recorded vibration exposure values. The resulting exposure values for the entire run were calculated by combining the data from all of the two-minute segments.

Since introducing the continuous method of recording acceleration, 23 seating locations have been recorded on 11 locomotives traveling 11 different routes across various parts of the United States. One of the routes was a shift of 'yard work', traveling back and forth in a rail yard coupling train cars together.

Results

A guide to interpreting weighted acceleration values with respect to health is given in Annex B of the 1997 ISO standard. A health guidance caution zone is defined to indicate the

level of vibration where a health risk could exist. The figure below shows the caution zone from the ISO standard as the area between the dashed lines. The dotted lines represents an alternative caution zone, also defined in the 1997 ISO standard, that is based on an estimated Vibration Dose Value (eVDV) and a health guidance caution zone range of 8.5 to 17 $m/s^{1.75}$. Also shown are the data points representing the exposure levels for all 23 measurements in all three directions. In all cases, the weighted rms accelerations measured were below both caution zones defined in ISO 2631.



To put the locomotive vibration exposure level in perspective, the results were compared to the levels measured on heavy trucks, light and medium duty trucks, a van and a motorcycle. The locomotive vibration levels were also compared to levels reported for various vehicles found in the literature. The vibration environment on locomotives was found to be comparable to commercial on-road vehicles and below many commercial off-road vehicles and recreational vehicles.

To evaluate the effect of transient vibration and shock, a VDV was calculated for all of the measurements. The VDVs calculated for locomotive crew members averaged 4.6 $m{\cdot}s^{-1.75}$, with the highest value at 6.1 $m{\cdot}s^{-1.75}$. These values are well below the action level of 15 $m{\cdot}s^{-1.75}$ defined in the British Standard (BS 6841:1987), the EU Directive 'action value' of 9.1 $m{\cdot}s^{-1.75}$, and the EU 'exposure limit' of 21 $m{\cdot}s^{-1.75}$.

Discussion

The vibration exposure experienced during locomotive operation was found to be consistently below the health guidance caution zones defined in the ISO whole body vibration exposure standard. The Vibration Dose Value measure of vibration exposure, which is an additional measure that is more sensitive to occasional shocks, was found to be less than the action levels of the British Standard and the EU Directive.

Environmental Effects on Truck Driver ISO 2631 Acceleration Exposure

Jack Wasserman, Logan Mullinix, Kelly Neal, Shekhar Khanal, Don Wasserman

Introduction

This paper presents current finding on truck driver average exposure to acceleration for several different manufacturer's cab-over trucks on a variety of roads in different countries. The predominant time, for this aspect of the study has been spent in the area around London, England and Warsaw, Poland.

The ECE directive 2002/44/EC has provided specific guidelines for vehicle operators 8 hour average acceleration exposure. The primary considerations have been on truck design including the air-ride driver's seat. The truck manufacturers have produced truck cabs that have some separate suspension from the truck frame. The truck seat manufacturers have been producing air-ride suspension seats for the cab. Both of these designs have had the objective of meeting the ECE directive and providing the vehicle drivers with some degrees of comfort.

This paper will provide some information on the ability of the vehicles to operate on a variety of roads and meet the objectives.

Method

The primary method for evaluation of the driver's exposure has been the use of a seat pad attached to the driver's seat. Although this sensor system provides the critical information for the driver, an understanding of the reasons for the values requires additional measurements.

The initial study in England used both driver and passenger seat pads, as shown in Figure 1, as well as triaxial accelerometers mounted on the base of the seat. The latest studies used significantly more transducers to better understand the relative rotations and translations on the truck frame, the cab, and the driver.

Figure 1 Triaxial Acceleration Seat Pad

The data was processed to produce the average accelerations for the X - axis, Y - axis, and Z – axis based on data for 360 seconds or longer. The time length is required by the ISO 2631 standard for reasonable accuracy.

Results

The data has shown road situations that have exceeded the 0.5 m/s^2 but not to exceed the 1.15 m/s^2 for extended periods of time. Comparisons between loaded and unloaded trucks and between different drivers have been done for certain situations. The major aspects related more to the road quality than the particular manufacturer for a vehicle or a seat. As can be seen in Figure 2, the driver's seat generally has lower values than the passenger's seat.

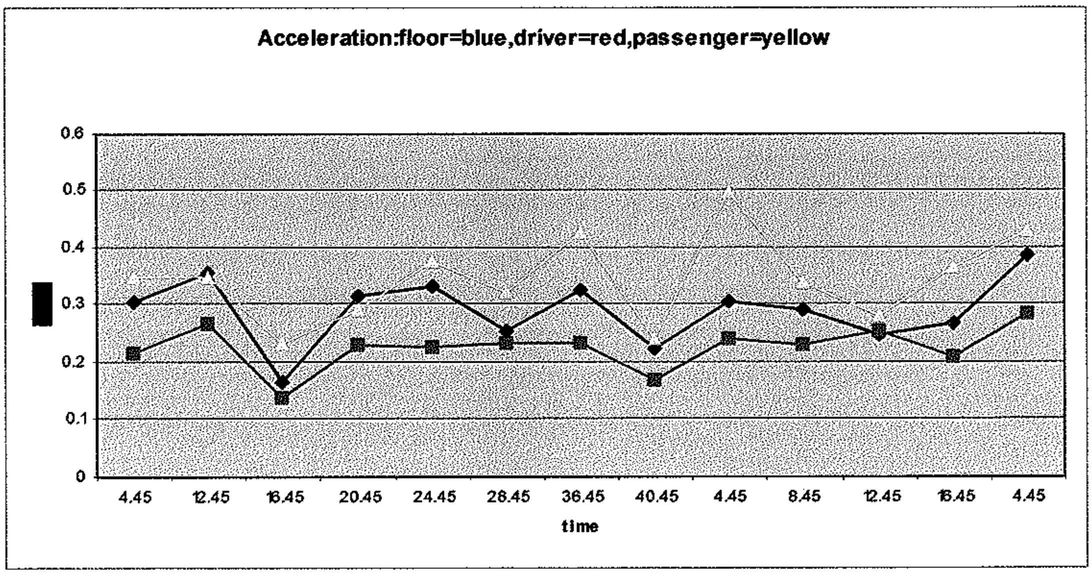


Figure 2 Comparison of Seating during time.

Conclusions

The initial results have shown that the dominant effects of the levels of acceleration expose have related to the quality of the roads and the truck speed. Continued testing is planned for the future to further understand the potential risks to the drivers and to allow a better process for assessment and design of truck seats.

EVALUATION OF THE CAPABILITY OF SEAT SUSPENSION TO REDUCE THE OPERATOR EXPOSURE TO VIBRATION IN TRACK TYPE TRACTORS.

Michael S. Contratto, Engineering Specialist, Caterpillar, Inc.
Tom Brodersen, Director - R and D, Sears Seating
Dave Marshall, R & D Manager, KAB Seating

Introduction

The European Union (EU) completed a new directive 2002/44/EC[1] called the Physical Agents Directive (PAD) that establishes action and limit values for hand-arm and whole body vibrations. The directive specifies that:

> " ...workers shall not be exposed above the exposure 'limit value'." [1]

and

> " ...once the exposure action values ... are exceeded, the employer shall establish and implement a programme of technical and/or organisational measures intended to reduce to a minimum exposure to mechanical vibration and the attendant risks..." [1]

The PAD limit value is effective for new machines starting July 6, 2007 and for used machines by at least July 6, 2010. These requirements apply to the users of machines, but machine manufacturers will be challenged to provide machines and information to help the users comply with the directive.

Caterpillar manufactures machines with the goal of enabling our customers to comply with all regulations dealing with health and safety. Caterpillar designs all of our machines to provide a safe, comfortable and productive work environment. This study was to determine if seat suspensions could provide a reduction in the vibration environment experienced by operators of Caterpillar mid sized (<50,000KG) Track Type Tractors

Methods

Seat manufacturers were asked to provide seat suspensions that provide improved isolation over and above current seat suspension. Each supplier was provided with ride profiles and was asked to demonstrate the vibration reduction on a shaker table. Two suppliers provided suspensions that were compared with the current seat suspension in a field study. Three full factorial experiments were conducted. The first experiment was to evaluate overall suspension performance for four operations. The second experiment was to determine the benefit of the adjustable vertical damper at three different levels and the third experiment was to determine the effect of the fore/aft and side/side isolators. Six operators were used for the study. Acceleration was measured at the seat base and at the operator seat pad. Both the transfer function and the ISO 2631 RMS ride values were used to determine the seat suspension's effectiveness of isolating the operator from vibration. A structured questionnaire was used to determine the operators' subjective assessment of the seat suspensions.

Results

There was significant operator-to-operator variability in the vertical direction (>35%), however there was little variability in the fore/aft and side/side direction based on seat base

acceleration. There was significant variation in the vertical vibration levels for all four operations; slot dozing, ripping, cross v-ditch, and roading. Roading showed much lower fore/aft and side/side vibration levels then the other operations. Slot dozing showed lower side/side vibration levels than other operations but was similar to roading. The fore/aft and side/side levels appear to be a function of the ground profile.

The seat suspensions demonstrated reductions in the ISO Ride values for the vertical direction in the shake table test however they did not show any significant reduction during the field operations. The exception was during the roading operations where the advance seat suspensions showed measurable reductions. The damper settings again showed significant differences during the shake test but had little or no effect during the field test. The fore/aft isolator did not provide a statistically significant reduction in the ISO ride values however the Side/Side isolator did provide a 20% reduction.

The seat suspensions did provide an improvement in the operator subjective evaluation of the machine vibration environment. In the vertical directions, the operators felt the advanced suspension provide a slight improvement. The fore/aft isolator provided a significant improvement in the vibration environment. This occurred despite the fact that the isolators provided no statistically significant improvement in the ISO ride values. The side/side isolator did provide a slight improvement in the operator perception of the vibration environment.

Discussion

Seat suspensions tested will not provide a significant reduction in the ISO RMS ride values for the current generation of construction machines however they do provide a significant improvement in the operator subjective opinion of the machine vibration environment. This may imply that the methodology used in the European Union (EU) directive 2002/44/EC may not be appropriate for evaluating operator comfort in construction machines. The basis of the ISO weighting curves are human response testing in a seated position without foot pedals, seat backs, arm rests and control contact. The operator seated position in construction machines may change how the human responds to vibration and perceives vibration. Further work is required to understand the effect of foot pedals, back rests, arm rest and control contact on the operator perception of the vibration environment.

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MUSCULOSKELETAL SYMPTOMS AMONG OPERATORS OF HEAVY MOBILE EQUIPMENT

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The purpose of this study was to assess the adequacy of the cab design and to determine the percentage of musculoskeletal symptoms among operators of mobile equipment used in mining and construction. A questionnaire was designed to assess demographics, work information, job history, and musculoskeletal symptoms in operators of heavy mobile equipment. Information concerning equipment included design of the seat/chair, levers, pedals, bothersome vibration, quality of ingress/egress from the equipment, proper preventative maintenance and repairs, and age of the equipment. The body regions that were evaluated included the neck, middle/upper back, low-back, shoulder/upper-arm, elbow/forearm, wrist/hand, hip, knee, and ankle/foot. Five hundred and eighty six operators completed the questionnaire. The results indicate that these workers are at risk for developing musculoskeletal disorders, and the need to quantify risk factors (i.e., whole-body vibration and static sitting postures).

Introduction

Kittusamy and Buchholz[1] estimated that there are currently 540,000 operators of heavy mobile equipment, who are generally referred to as operating engineers, in the United States. Their estimate also shows that ninety percent of the operating engineers are involved in performing excavating and paving work, whereas the remaining 10% are crane operators and all of these operating engineers are exposed to whole body vibration. Two important risk factors for musculoskeletal disorders among operators of heavy earth-moving equipment are static sitting and whole body vibration,[2,3] where long term exposure to these risk factors have been associated with low back pain, disc degeneration, sciatic pain, and muscle fatigue.[4]

Methods

A work and health questionnaire was designed to assess demographics, work information, job history and musculoskeletal symptoms in operators of heavy mobile equipment. Self-administered work and health questionnaires were distributed to operating engineers by the International Union of Operating Engineers training centers in several states within the United States of America. The operators who attended their regularly scheduled training classes, from December 2001 to May 2005, at the training centers were requested to complete the questionnaire during their training session. The participation was voluntary, but participation was highly encouraged by the training officers. All of the participants were briefed about the purpose of the study and they signed an informed consent form.

Results

Five hundred and eighty six operators out 598 (98%) completed the questionnaire from 6 different local unions in 8 different states. A majority of the participants were male (91%). A majority of the operators (72%) were journey level. The ages of the operators ranged from 18 to 68 years. The majority of the operators (>65%) indicated that the cab (i.e., seat/chair, levers and pedals) was adequately designed for their job. Some of the operators reported that they were not bothered by vibration and that the quality of egress from the equipment was good. Most of the operators (>80%) indicated that proper maintenance and repairs were performed on their equipment. The classification of equipment as being old or new was almost identical.

The prevalence of musculoskeletal symptoms in the total population was 58.5%. Three body regions that received the highest total percent of symptoms categorized as somewhat severe or higher, included the knee, shoulder/upper-arm, and the low back.

Summary

The current study is in agreement with the prevalence of musculoskeletal symptoms in various body regions as reported by Zimmerman et al.(5) Also, similar results were observed in a pilot study of operators of heavy construction equipment that further reiterate the findings in the current study(6) .

Construction workers are often afflicted with musculoskeletal symptoms that compromise their health and well-being. However, there have been few formal studies of the nature and potentially preventable causes of these symptoms. The results from this study indicate that the operators are at risk for developing musculoskeletal disorders, the need to quantify risk factors (i.e., whole-body vibration and static sitting postures), and develop engineering controls to reduce the exposure levels.

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Poster Presentations

HEAD-TRUNK MOTION INCREASE WITH ARM-REST CONTROLS

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Introduction

Heavy equipment manufacturers have made a long-term commitment to minimize operator vibration exposure for comfort, performance, and health reasons. Domestic and international guidelines/standards and EC laws dictate exposure limits based on measurement of vibration at the interface between the seat and the operator's buttocks using seat-pad accelerometry.[1-4] This is historically based on the assumption that the only major source of vibration is transmitted through the seat pan. However, vibration may also be imparted to the head and neck via the steering wheel and/or arm-rest controls and a relatively rigid upper body.[5] Unfortunately, little is known regarding the influence of arm position on head and neck motion. The purpose of this study was to investigate relative head and trunk motions during riding simulations of large construction equipment, using three different arm control options.

Methods

Five typical heavy equipment ride files were "played back" through a man-rated Servo Test 6-degree-of-freedom vibration system. An 8-camera Vicon motion capture system operating at 200 frames per second, recorded the motion of reflective surface markers on 5th, 50th, and 95th percentile right-handed male subjects, using 3 seat and control configurations (steering wheel (SW), floor mounted armrest controls (FM), seat-mounted armrest controls (SM)). Two trials were performed for each ride and seat control combination (each trial: 60 sec of 6-dof and 60 sec of vertical vibration). The relative motions (change in distances) from the marker over the xiphoid process (caudal end of sternum) to markers over each shoulder, each mid-clavicle, the presternal notch, and to each of four markers on a tight band around the head were calculated (12,001 frames, 6-dof motion only). As a rigid body control, distances between markers on the head band were also monitored. The standard deviation (SD) of the 12,001 distances between pairs of markers was normalized by the mean (L) of the associated distances producing: SD/L which was used as a measure of motion. Error assessments were also performed by analyzing the motion between relatively fixed markers (on the headband). A repeated measures analysis of variance was used to evaluate the results. While five ride files were used, only one ride file containing significant lateral acceleration components was analyzed for comparing the effects of two armrest controls versus use of a steering wheel for this part of our study.

Results

Values of SD/L between the points on the relatively rigid head band were consistently small and similar to each other for all conditions with one exception due to treatment (SM v SW, $p=0.0145$). SD/L between the markers over the xiphoid process and the presternal notch, another region that should be relatively rigid, were also similar to each other for all conditions. Use of floor-mounted, arm rest controls versus a steering wheel produced a significant increase in the value of SD/L between the xiphoid process and: the right shoulder marker (92%, $p=0.0316$), the right mid-clavicle marker (47%, $p=0.0478$), and the right-front marker on the head band (28%, $p=0.0182$). Use of floor-mounted, arm rest controls versus seat-mounted, arm rest controls

produced a significant increase in the value of SD/L between the xiphoid process and the right-back marker on the head band (14%, p=0.0467).

Discussion

During a pilot study to assess the efficacy of a motion capture system in whole-body vibration studies, the authors observed a large increase in head-trunk relative motion due to the use of armrest controls, raising a concern about an increased likelihood of injuries. With the use of a steering wheel, the trunk and arms can behave as active dampers, attenuating horizontal motions and maintaining a stable platform for the head-neck system (an inverted pendulum). Armrest controls more rigidly couple the shoulders, via the upper arms, to a vibration source and bypass the damping provided by the entire arm, potentially increasing the risk of motion-related musculoskeletal problems in the neck and upper trunk. While armrests may reduce arm and shoulder fatigue and reduce the effect of the vibrating trunk mass on the lower back, they may do so at the expense of increased motion at the neck and shoulders. The vibration community needs to consider the effect of and attenuation of vibration from sources other than the seat pan. The authors urge the standards and law making communities to consider vibration sources in addition to those at the operator's seat pan.

Acknowledgements

This study was funded by Caterpillar, Inc. of Peoria, IL and was conducted at Sears Manufacturing of Davenport, IA with the help of Mike Drinkall and Jason Boldt. Dean Macken of the Engineering Design and Prototyping Center at the College of Engineering at The University of Iowa helped a great deal with design and setup of specialized equipment. Brad Parker, of the Center for Computer-Aided Design at The University of Iowa provided vital help with computer hardware and software. The study was conducted through the Center for Computer-Aided Design (directed by Karim Abdel-Malek at The University of Iowa), where one of its goals is to optimize the Digital Human.

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ARM AND SHOULDER MUSCLE ACTIVITY ARE GREATER WITH STEERING WHEEL VS. SEAT MOUNTED CONTROLS

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Introduction

Chronic whole-body vibration exposure, as expected in large construction and mining vehicles, has been associated with neck and back pain and injury.[1] While work has been done towards gaining a better understanding of the relationship between vibration and shock and muscle activity of the back musculature[2], relatively little information regarding the activity of neck, shoulder and upper arm muscles is known. Today's equipment designs must conform to domestic and international standards, however these standards do not specifically address the vibration exposure in the head and upper quarter. Further it is not well known how the control configuration within a vehicle (e.g. steering wheel versus arm controls) influences muscle voluntary and reflex activity levels. Greater muscle activity may lead to greater muscle fatigue – which in turn may be associated with greater risk of injury.[2] Thus, muscle contractions needed to maintain static postures as well as those resulting reflexively should be considered during an analysis of seating position. Unfortunately, little is known regarding the influence of arm position on head and neck muscle function. The purpose of this study was to investigate the relative muscle activities of 5 neck, shoulder, and upper arm muscles during riding simulations of large construction equipment, using three different arm control options.

Methods

Five typical heavy equipment ride files were "played back" through a man-rated Servo Test 6-degree of freedom (dof) vibration system. Each ride was repeated using 3 seat and control configurations (steering wheel (SW), floor mounted arm-rest controls (FM), seat mounted arm-rest controls (SM)). Two trials were performed for each ride and seat control combination (each trial: 60 sec of 6-dof and 60 sec of vertical vibration). Five channels of surface electromyography (EMG) of the right-side cervical erector spinae muscles (neck extensors), sternocleidomastoid (neck flexor), upper trapezius (shoulder elevator), biceps brachii (elbow flexor) and triceps brachii (elbow extensor) muscles were collected throughout each ride (~ 2min) using pre-amplified (10x), 1 cm silver bar electrodes, with 1 cm fixed inter-electrode distances (Delsys, Inc). Further analog amplification was set at 10k (1k for one subject), and sampled at 1000Hz using a 12-bit DAQ card and Labview 7.1 software (National Instruments). A total of 7 right-handed male subjects were tested, but only 5 had complete EMG data sets to analyze for this sub-study. EMG was analyzed using root mean square (RMS, in mV) of 20 ms moving windows, and then averaged across the entire trial for a measure of mean total muscle activity (voluntary and reflexive). The muscle activity to maintain the static posture was estimated as the mean RMS EMG over a 1 sec interval just prior to and/or after completion of the ride. Repeated Measures ANOVAs were used to test for with-in subject differences using $\alpha = 0.05$.

Results

The upper trapezius and triceps brachii muscles were significantly more active (mean EMG muscle activity) while using the steering wheel controls than for either the floor mounted or seat mounted arm rest controls. Whereas, the floor mounted arm controls tended to produce greater activity in the biceps brachii. Overall, the seat mounted controls resulted in the lowest mean EMG levels across all five muscles. No significant differences were observed in the neck flexor (sternocleidomastoid) or the neck extensor (erector spinae) muscles across control configurations.

Discussion

This pilot study suggests that muscle activity is indeed influenced by arm control postures. In our companion study on relative neck and shoulder motion, we indicate greater relative motion with the armrest control configurations. Interestingly, in this study we observed greater static and dynamic muscle activity with the steering wheel configuration. The arms may behave as active dampers particularly when the control configuration is not mounted to the seat (SW or FM), potentially attenuating head and neck motions. However, it is not entirely clear as to whether the greater relative motion or the potential for greater muscle fatigue over time may be the most problematic for equipment operators. Certainly the risk of injury may depend on the type of injury considered, e.g. overuse muscle injury versus repetitive motion joint pathology. There may be trade-offs between the potential for reduced fatigue associated with arm-rest controls, which is supported by our observations of decreased mean muscle activity, and the potential for greater apparent muscle and joint stiffness associated with tonic muscle activity – and thereby reduced motions. These preliminary results would suggest that the vibration community needs to consider the effect of and attenuation of vibration in the upper quarter considering the influence of postural muscle activity with different arm control configurations on the transmissibility of vibration into the head and neck.

Acknowledgements

This study was funded by Caterpillar, Inc. of Peoria, IL and was conducted at Sears Manufacturing of Davenport, IA with the help of Mike Drinkall and Jason Boldt. At the University of Iowa, Dean Macken of the Engineering Design and Prototyping Center in the College of Engineering assisted with design and setup of specialized equipment, and Brad Parker, of the Center for Computer-Aided Design (CCAD) provided vital help with computer hardware and software. The study was conducted through CCAD (directed by Karim Abdel-Malek) where one of its goals is to optimize the Digital Human.

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EVALUATION OF POWERED WHEELCHAIRS WITH SUSPENSION AND EXPOSURE TO WHOLE-BODY VIBRATION

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Introduction

Although wheelchair users are regularly subjected to whole-body vibrations little research has been conducted to assess these vibrations or attempt to reduce them [2,3,5]. Most of the wheelchair and whole-body vibration research done to this point has been conducted on manual wheelchairs. Van Sickle et al showed that manual wheelchair propulsion over a simulated road course produces vibration loads that exceed the ISO 2631-1 standards for the fatigue-decreased proficiency boundary at the seat of the wheelchair as well as the head of the user [6]. In a study by Boninger et al [1], 66% of wheelchair users reported neck pain since acquiring their wheelchair. One of the key reasons believed to be the cause of pain, was the exposure to whole-body vibration. Kwarciak et al [4] and Wolf et al [7] performed similar studies using two methods of analysis to evaluate vibrations on suspension and non-suspension wheelchairs while descending curbs of varying heights. Both studies revealed no significant difference in the abilities of the wheelchairs to reduce the amounts of vibrations transferred to the wheelchair user. Although the efforts of wheelchair companies to reduce the amounts of whole-body vibration transmitted to wheelchair users through the addition of suspension systems is encouraging, the technology is not yet ideal. Additionally, the research to date has focused on manual wheelchairs exclusively, while little attention has been shown to powered wheelchairs.

Methods

This study includes the use of two suspension electric powered wheelchairs: The Quickie S-626 and the Invacare 3G Torque SP Storm Series. Each subject tested all of the configurations of the suspension wheelchairs. These included the Invacare with suspension, the Quickie with suspension set to three settings (most stiff, least stiff, and 50% stiffness), and both wheelchairs with solid inserts to act as non-suspension wheelchairs. Sixteen able bodied subjects have been recruited for this study so far. In each of the configurations of the wheelchairs, the subjects traversed an Activities of Daily Living (ADL) course. Vibrations were collected from a tri-axial accelerometer attached to a seat plate beneath the cushion during driving over the activities course. A mixed model ANOVA was used to determine if there were differences between suspensions based on Vibration Dose Value (VDV).

Results

Statistical analyses of the VDV data revealed significant differences between the six different suspensions over each of the obstacles in the activities of daily living course. Post-hoc analyses revealed that for each of the obstacles, significant differences existed between the Invacare suspension and the Invacare solid insert. For the Quickie power wheelchair the solid insert setting was not significantly different from the most-stiff setting for each of the obstacles

except the smooth surface. The solid insert setting was significantly different than the lowest and mid stiffness settings for all of the obstacles except the smooth surface and the deck surface.

Table 1 – Average and total VDV values ($m/s^{1.75}$) for each suspension setting

	Invacare Insert	Invacare Suspension	Quickie Insert	Quickie Least Stiff	Quickie Mid-Stiff	Quickie Most Stiff
Deck	0.23	0.26	0.25	0.23	0.23	0.25
Door	1.07	0.72	0.81	0.56	0.51	0.77
Curb	2.45	1.56	2.87	1.41	2.06	2.78
Dimple	0.69	0.61	0.69	0.59	0.58	0.68
Smooth	0.14	0.11	0.14	0.12	0.12	0.15
Carpet	1.00	0.83	1.16	0.71	0.70	1.02
Total VDV	**2.55**	**1.65**	**2.91**	**1.55**	**2.10**	**2.87**

Discussion

Although most of the suspension systems are capable of reducing the amounts of vibration transmitted to the users, the exception being the Quickie S-626 with the most-stiff suspension setting (this setting was not significantly different from the solid insert setting for all obstacles except the smooth surface), the results of the vibration dose values seem to indicate that they may not reduce them enough to reduce probability of injury in powered wheelchair users. When examining the total VDV over the entire activities of daily living course, in relation to the Health Guidance Caution Zone (HGCZ), there is not significant time allowed before WBVs are considered dangerous.

The information on the transmissions of vibrations from different suspension systems can lead to improvement in their design and function allowing powered wheelchairs to adequately reduce the amount of whole-body vibrations experienced by their users. Future research should investigate vibrations experienced by wheelchair users in real environments over extended periods of time.

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ESTABLISHMENT OF AN EXPERIMENTAL SYSTEM FOR MEASRING BIODYNAMIC RESPONSE OF HAND-ARM

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Introduction

This paper addresses establishment of an experimental system for measuring biodynamic response (BR) of hand-arm system at the NIIH in Japan. BR measurement system at the NIIH is nearly equivalent to NIOSH installed system. The feasibility of the system is examined through the apparent mass (AM) measurement of the empty handle and a set of calibration masses.

Apparatus

The grip force was measured by using the handle shown in Fig. 1. The handle has two force sensors (KISTLER, 9212) and one accelerometer (PCB, 356A12). A low-pass filter with 5 Hz cut-off frequency was used to the grip force from measured force signal. Figure 2 shows BR measurement system in this study. The push or pull force at the handle was measured by using the force plate (KISTLER, 9286AA). The grip force and the push / pull force were displayed on a monitor. The shaker (IMV, VE-100S) is used to vibrate the hand-arm system along the forearm axis (Z_h direction) (ISO 10068, 1998; ISO 5349-1, 2001). In most situations force actions for operating tools are expressed by grip, push, pull and combined these actions. These actions can be simulated in the test system. AM was obtained by performing H1 estimator in the PULSE™ system (B&K, 3109) and it is denoted at the one-third octave band center frequencies.

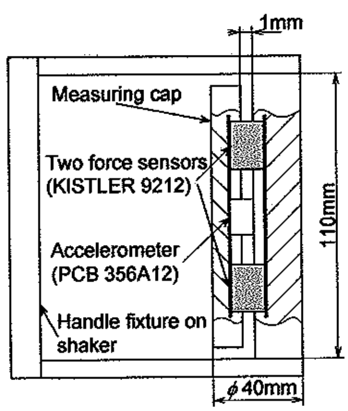

Fig. 1 Instrumented handle of the system

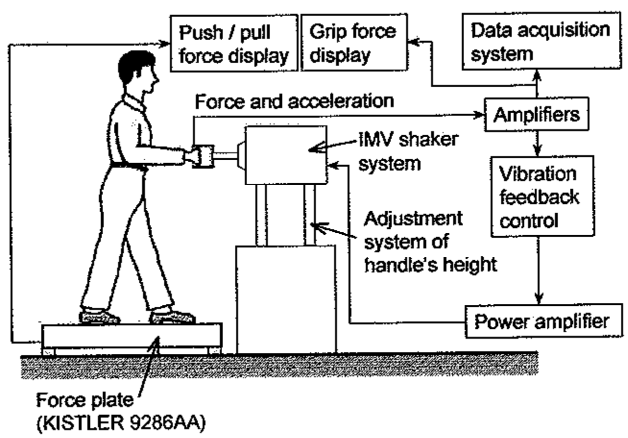

Fig. 2 Measurement system at the NIIH

Methods

In order to investigate the reliability of the system, AM measurement of the handle was performed. It is assumed that the handle is rigid in the upper limit of adoptive frequency range in this study. This assumption is validated in AM measurement of the empty handle. A pseudo-random vibration in the frequency range of 10 to 1,250 Hz was used and its amplitude is 1.0 $(m/s^2)^2/Hz$ with a flat power spectral density (PSD) in the experiment.

Measured AM includes the mass effect of the measuring cap in a subject experiment. Compensated apparent mass $AM_c(\omega)$ is obtained by Eq. (1) [1-2].

$$AM_c(\omega) = AM_{total}(\omega) - AM_{cap}(\omega) \quad (1)$$

where $AM_{total}(\omega)$ is measured response with the mass of the measuring cap and BR of a subject, $AM_{cap}(\omega)$ is the response of measuring cap in an empty handle test. In this study it is assumed that $AM_{total}(\omega)$ is the response with attached small piece of metal to the measuring cap by adhesive tape. Eight pieces (1, 2, 3, 4, 5, 10, 15 and 20g) of metal were used in the experiment.

Results and Discussions

The measured AM of the empty AM handle differences between measured the and true values are less than 3%. Since than resonant frequency is higher enough 1,000 frequency range of measurement (12.5 – Hz), the assumptions seem to hold in the frequency range of measurement. The calibrations of the measuring cap's mass are shown in Fig. 3. The measured pieces of mass generally agree with the true mass value. The measured mass values of over 10g are bigger than the true mass value in the high frequency range (>600Hz).

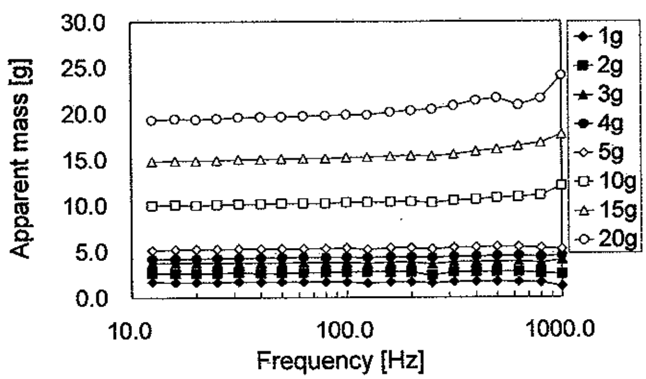

Fig. 3 Mass compensation results

The amplification of the response seems increases with the increase in the metal mass. This is likely because each piece of metal is resiliently attached to the measuring cap by adhesive tape and the metal and tape form a local 1D system. The resonant frequency of the system reduces with the increase in the mass value. This further supports the validity of the measurement system and the mass cancellation method.

Conclusions

Throughout the course of this study, several conclusions are obtained as follows:

(1) A BR experimental system for measuring biodynamic response of hand-arm system and vibration exposure tests was established in NIIH.

(2) The instrumented handle of the system was validated through the AM measurement.

(3) The mass of the measuring cap in the AM measurement was well compensated by the mass cancellation method, which confirms its validity.

Acknowledgements

The authors acknowledge the assistance of staff at NIOSH, Dr. Dong, R. G. and Mr. Welcome, D. E. Their help is greatly appreciated.

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Training Simulators Extend Laboratory Testing Techniques for WBV Analysis

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Shekar Khanal
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Don Wasserman

Introduction

Human testing has always been a needed way to provide information on the effects of vehicle vibration, however, the manner of testing has not reflected the real situations of driver's hands on a steering wheel and a seat with back support and driving tasks. The typical system have used a standard sinusoidal excitation rather than the typical types of road – truck excitations

The new truck driver training simulators provide the combination of road roughness, speed effects, cab environment and individual tasks. The system has a full six axis simulation potential. The simulators have the protection of the individual by a combination of two ways for the individual to stop the motion as well as an operator with visual capability who can stop the testing. The closed simulator, shown in Figure 1, has the potential for providing motion during the operation.

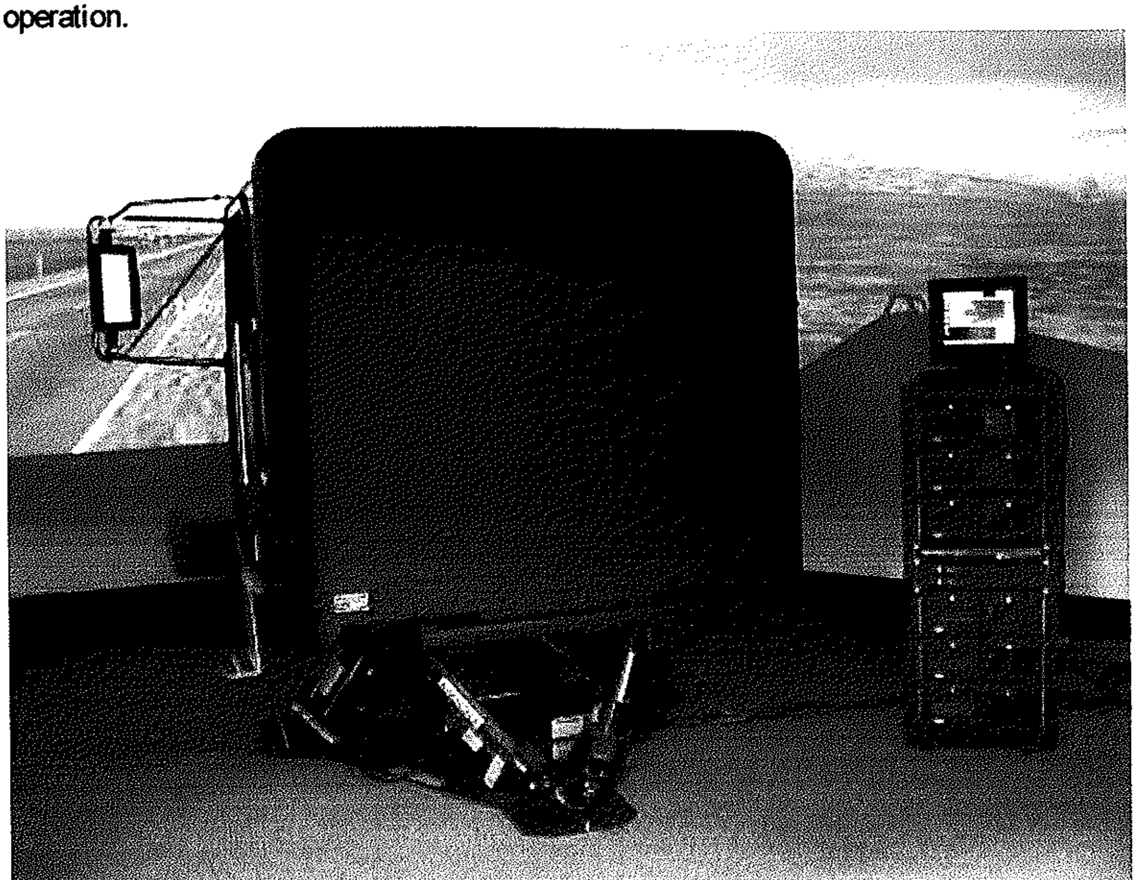

Figure 1 Mark III Truck Simulator

Plan Objectives

The current project is to evaluate the levels and distribution available from a standard truck driver training simulator. The simulator has a combination of regular routs and "rough" routs.

The system will be operated with a combination of triaxial seat pads and floor accelerometers for comparison to the date collected from the trucks in Europe

Results

Comparisons of the truck testing data will be provided as part of the planning for future research activity. Initial testing has been done on the vibration exposure for the operator of the simulator when the roads are "rough". The actual rms weighted value for vertical acceleration was 0.254 m/s^2. The 1/3 Octave spectrum shown in Figure 2 is from driver's seat in England. This seat showed significant loading in the 4 Hz. band. The simulator does have some loading in this area, but it is much lower.

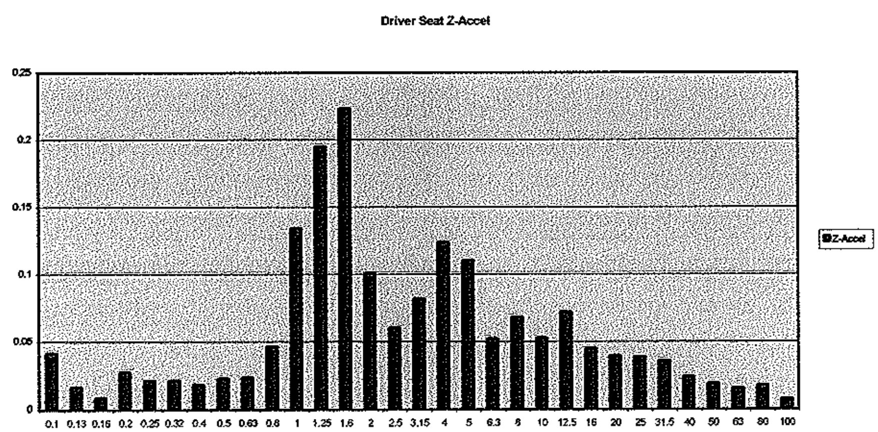

Figure 2 Driver's Seat 1/3 Octave Z –Axis Acceleration

For testing purposes, the values in the 4 – 8 Hz region may need to be increased to the normal band level.

INSTRUMENTED HANDLES FOR STUDYING HAND-TRANSMITTED VIBRATION EXPOSURE

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Introduction

Instrumented handles or dynamometers are widely used to measure hand forces and/or the biodynamic response of hand-arm system. To study hand-transmitted vibration exposure, six generations of instrument handles were constructed or initially developed by researchers in ECTB/HELD/NIOSH. This presentation provided a summary of these handles. Their basic characteristics, limitations, and usefulness are described, which may help their appropriate applications and further improvements.

Six Designs of Instrumented Handles

Handle 1: The conceptual design is recommended in ISO 10819 (1996)[1] for glove test. The grip force is measured by detecting bending strains on a measuring beam in the handle. A special handle fixture was designed to connect the handle to a shaker. Except the screws, the handle and fixture were made from aluminum.

Handle 2: The design is based on the principle of shear strain measurement.[2-3] Both grip and push forces can be measured simultaneously using this handle. This handle was directly designed for a simulated vibrating tool.

Handle 3: This design is basically composed of a handle base, a measuring cap, and two charge-based sensors (Kistler 9212) sandwiched between the base and cap. The handle was also made from aluminum. The fixture for Handle 1 was also used with this handle. This generation of handle has three different handle diameters (30, 40, and 50mm).[4]

Handle 4: This design is an improvement from Handle 3. The handle fixture was totally redesigned and it was much stiffer than the previous one. The aluminum measuring cap was replaced with a magnesium cap.

Handle 5: The basic structure of this handle is the same as that for Handle 3. However, the piezoelectric sensors were replaced with two strain gage based sensors (Interface SML-50).

Handle 6: This handle includes two measuring caps, four piezoelectric sensors, and a handle centre base. The handle fixture was the same as that with Handle 4.

Methods for Handle Examinations

The static and dynamic characterizations were performed using the methods reported by Dong et al.[5-6]

Results and Discussion

Handle 1: The static force measurement depended on the hand grip location on the handle. Its natural frequency was less than 200 Hz.[5,6] Because the transmissibility of gloves may not vary significantly with the applied grip force, this handle may be acceptable for glove test. However, the force measurements with this handle may not be reliable.

Handle 2: The static force measured with this handle was insensitive to the hand acting location. However, when the handle was vibrating, the force signals could be totally distorted. For this reason, it was not used for vibration studies.

Handles 3 and 4: The static force measurements with these handles were independent of the hand grip location.[6] The resonant frequency of the early version was about 1,450 Hz and the latest was about 1,900 Hz. These handles have been extensively used for both static and biodynamic measurement up to 1,000 Hz.[4,6] The experimental data measured with the handle have been used to develop biodynamic models. A sample model, together with its parameters, is shown in Fig. 1. The modelling results agree excellently with the experimental data, as shown in Fig. 2. The natural frequencies (29 Hz and 208 Hz), the damping ratios (0.29 and 0.73), and the potential static deformations of the hand-arm system in the possible hand force range are also very reasonable. Without the reliable and accurate experimental data, it is impossible to establish such a model.

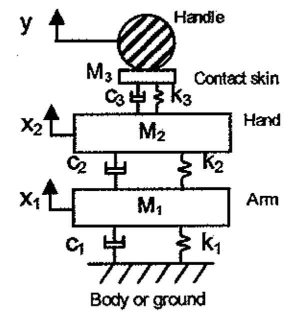

Fig. 1: A 3-DOF model (M_1=1.2320 kg; M_2=0.1774 kg; M_3=0.0338 kg; k_1 = 1.5 kN/m; k_2 = 48.5 kN/m; k_3=252.8 kN/m; c_1=54 N-s/m; c_2=104 N-s/m; c_3=231 N-s/m.)

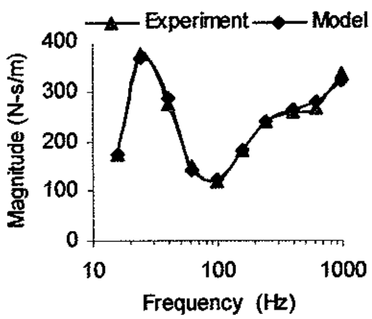

Fig. 2: Comparison of modeling and experimental impedance data (50 N grip-only) (r = 0.993).

Handle 5: Piezoelectric force sensor can have a significant zero-drift problem. The handle equipped with such a sensor may not be suitable for a long duration force measurement. The handle equipped with strain gauge sensors has no such a problem. However, because the sensor is not as stiff as the charge-based sensor, the handle resonance was at about 900 Hz. It has been used for studying hand force recall.[7]

Handle 6: Except for Handles 2 and 6, the other handles cannot simultaneously measure both grip and push forces. The push force is usually measured using a force plate in the experiment. The dynamic responses distributed on the fingers and palm can only be measured separately using Handles 3-5. Handle 6 was developed to overcome the deficiencies. Its natural frequency was about 1,450 Hz.

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A NOVEL THEORY: ELLIPSE OF GRIP FORCE

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Introduction

Hand forces are important factors for risk assessment of hand-arm vibration syndrome (HAVS).[1] Grip force is one of the most important force components in the operation of powered hand tools. A considerable number of studies on grip force have been reported. It is well known that the grip force applied on a cylindrical handle is not uniformly distributed on each axis across the center of the handle cross-section.[2] Therefore, maximum and minimum orientations of grip force exist around the handle. Such orientations have not been clearly identified. In a recently proposed international standard (ISO/CD 15230, 2005),[3] it is stated that "the direction of the main gripping force is generally parallel to the z-axis defined in ISO 8727." [4] This assertion is questionable, and further examinations are required. The objective of this study was to establish a fundamental theory on the distribution of the grip force around cylindrical handles.

Methods

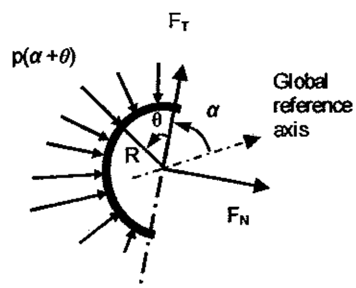

Fig. 1 Grip force definition

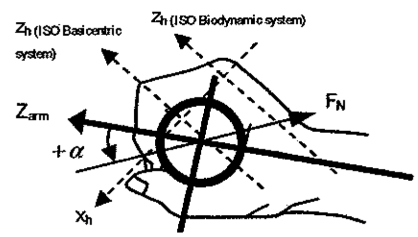

Fig. 2: Hand coordinate systems (z_h & x_h: ISO system.[1,4] Z_{arm}: forearm z-axis. F_N: grip normal force at α deg.

As shown in Fig. 1, the grip force is defined as a vector composed of normal and shear components (F_N & F_T) in this study. Fig. 2 shows the hand coordinate system defined in this study, together with the ISO systems.[1,4] Based on this novel grip force definition, we derived four fundamental properties and a theorem. More significantly, we formulated a novel hypothesis: similar to Mohr's circle of stress, the normal and shear components can be represented approximately using an ellipse on the plane of the two force coordinates, as shown in Fig. 3. The maximum and minimum grip normal forces are termed as the first principal grip force (F_1) and the second principal grip normal force (F_2), respectively. Their corresponding orientations are termed as the first principal grip angle (α_1) and the second principal grip angle (α_2).

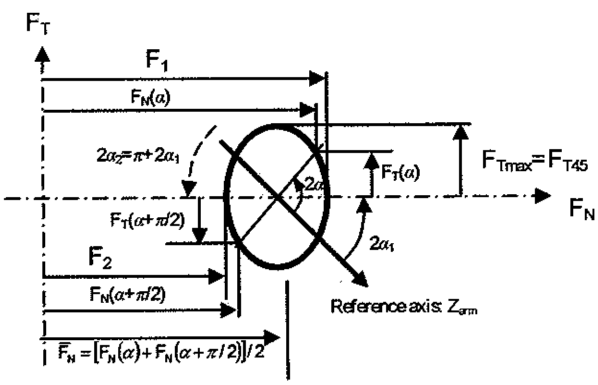

Fig. 3: A novel hypothesis: ellipse of grip force.

A series of experiments were conducted to test this hypothesis. Twelve

subjects participated in the experiment. Three cylindrical handles (30, 40, and 48 mm) were used. Each of them was equipped with a flexible contact pressure sensor (TekScan, Model #5101-100). Fig. 4 shows the measurement setup and hand grip posture. Each subject was required to align the hand mark (on Z_{arm} axis) with the handle mark and to apply the maximum and medium (50%) grip forces on the handle.

Fig. 4: Test setup & hand posture

Results and Discussion

Fig. 5 shows an example of the experimental results. Table 1 provides comparisons of the elliptical model predictions and the test data. The results strongly support the hypothesis. This study also found that the maximum grip pressure around the handle is distributed in the finger contact area. On the 40 mm handle, the first principal force is more than 40% of the second principal force (t-test: $p < 0.001$). The maximum force is located in the finger contact orientation at approximately 27° from the Z_{arm}-axis that is about 29° from the hand z_h-axis defined in ISO 5349-1 or ISO 8727. It is significantly greater than that on the Z_{arm}-axis (t-test: $p < 0.001$). The maximum force on the 30 mm handle moves further from the Z_{arm}-axis, and that on the 48 mm handle moves closer to this axis. Therefore, even if the z_h-axis in the basicentric system defined in ISO 8727 could align with the Z_{arm}-axis in the operations of some tools, the above-mentioned statement in ISO/DIS 15230 (2005) is generally invalid. The proposed theory can be used to improve the standard and to develop a more effective method for grip force measurement.

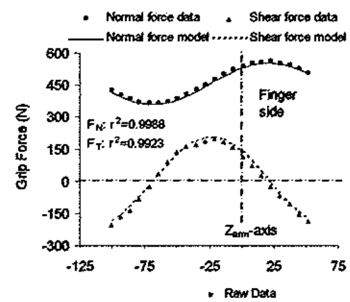

Fig. 5: Data comparisons

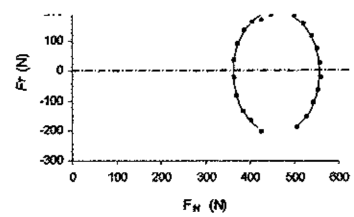

Table 1: Modelling and test data for 40 mm handle

Ellipse Parameters	α_1- α_2 (deg.)	F_{T45} $/F_{Tmax}$	$(F_1$-$F_2)$ $/F_{Tmax}$	r^2-value for F_N fitting	r^2-value for F_T fitting
Mean	89.4	0.9741	0.9313	0.9937	0.9642
SD	5.8	0.0286	0.0767	0.0082	0.0425
Theory	90.0	1.0000		1.0000	1.0000

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CHEST TRANSMISSIBILITY CHARACTERISTICS DURING EXPOSURE TO SINGLE- AND COMBINED-AXIS VIBRATION

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Introduction

Ground, air, and water vehicles can expose humans to substantial multi-axis vibration. Multiple input/multiple output relationships or models exist for estimating frequency response functions of linear systems[1, 2]. These relationships have been applied by some investigators to evaluate the effects of occupied seat vibration[3, 4]. Using a multiple input/single output model, this study investigated the effects of single- and combined-axis vibration in the fore-and-aft (X), lateral (Y), and vertical (Z) directions on vibration transmission to the human chest. Frequency response functions (transmissibilities) were estimated and compared for the back-on and back-off postures.

Methods

A rigid seat with seat back was mounted onto the Six Degree-of-Freedom Motion Simulator (SIXMODE). A flat acceleration vibration signal was generated between 2 and 40 Hz at 1.0 ms^{-2} rms in the single and combined X, Y, Z, XY, XZ, YZ, and XYZ axes. The signals were shifted in time so that the combined inputs were not fully correlated. Lightweight triaxial accelerometers were used to measure accelerations at the seat base (input) and at the bony manubrium of the chest (output). The maximum of nine frequency response functions ($H(\omega)$) or transmissibilities were estimated from the auto- and cross-spectra. The system transfer matrix for the XYZ inputs and chest Z output is

$$\begin{bmatrix} H_{xZ} \\ H_{yZ} \\ H_{zZ} \end{bmatrix} = \begin{bmatrix} P_{xx} & P_{xy} & P_{xz} \\ P_{yx} & P_{yy} & P_{yz} \\ P_{zx} & P_{zy} & P_{zz} \end{bmatrix}^{-1} \begin{bmatrix} P_{xZ} \\ P_{yZ} \\ P_{zZ} \end{bmatrix} \qquad (1)$$

where P_{xZ}, P_{yZ}, and P_{zZ} are the cross-spectra between the three inputs at the seat base and the Z output at the chest, respectively, and P_{xx}, $P_{xy,\ldots zz}$ are the auto- and cross-spectra between the input signals (ω not shown in Eq. 1). Equation 1 can be similarly written for the chest X and Y outputs. Matlab® was used to estimate the auto- and cross-spectral densities for calculating the transmissibilities, ordinary coherences (for single inputs), partial coherences, and multiple coherences.

Results

Figure 1 illustrates the major chest transmissibilities observed for the two postures. Vertical vibration showed a consistent influence on the chest X response (Chest X/Z), most likely causing chest pitch. Some chest Z responses were observed with X-axis inputs, but the results were variable and difficult to interpret. In general, other factors besides the known inputs did not affect the transmissibilities shown in Figure 1 (Repeated Measures ANOVA, P<0.05). This was

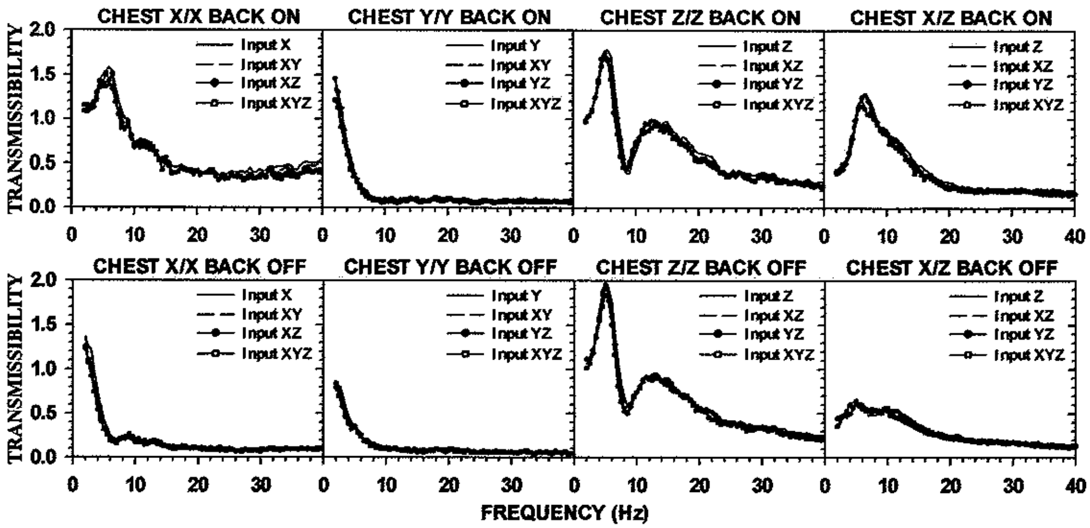

Figure 1 Mean Chest Transmissibilities from Nine Subjects (4 Females, 5 Males)

reflected by the relatively high partial coherences, particularly associated with the primary peak responses (majority PCoh>0.85). More variable coherences were noted among the subjects for Chest X/Z for the XZ and XYZ inputs, the lowest mean value being 0.75±0.14. Regardless of the input, the back-off posture showed the elimination of the 4-6 Hz peak in Chest X/X, the significant reduction in the peak frequency for Chest X/Z, and the significant reductions in the Chest Y/Y and Chest X/Z transmissibilities (Fig. 1, Paired t-test, $P<0.05$).

Discussion

Lower partial coherences would suggest that the chest responses were not fully accounted for by a linear relationship to the known inputs. This could occur due to chest pitch, which was expected to some extent with both the X and Z inputs. Except for a few cases, the partial coherences were relatively high. The seating posture was found to have a significant effect on the chest multi-axis biodynamics. Specifically, coupling with the seat back promoted the influence of vertical vibration on the chest X response, causing higher upper torso motion in the X direction at a peak coincident with whole-body resonance (~4-6 Hz, as observed in Chest Z/Z). When contact with the seat back was removed, these effects were reduced and the peak chest X motions appeared dampened at higher frequencies. The chest X motion with the back off appeared to be more influenced by lower frequency vibration associated with relatively higher seat displacement (~2 Hz).

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A FIELD STUDY: MEASUREMENT AND EVALUATION OF WHOLE BODY VIBRATION FOR MH-60S PILOTS

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Introduction

Pilots of the MH-60S helicopter are exposed to continuous whole body vibration (WBV). Pilot fatigue is a growing operational concern due to the increased frequency of extended durations of missions (6-8+hours) in support of Operations Iraqi Freedom and Enduring Freedom. Endurance aspects of the currently used rotary wing seating systems were not optimized for the longer missions and wide range of pilot anthropometric measurements, which is now typical of naval aviation. The current seating systems were designed primarily to meet crashworthiness requirements, not for the wide range of pilot anthropometry or to mitigate WBV. Albeit, an issue, pilot fatigue and reduced mission effectiveness are also critical concerns.

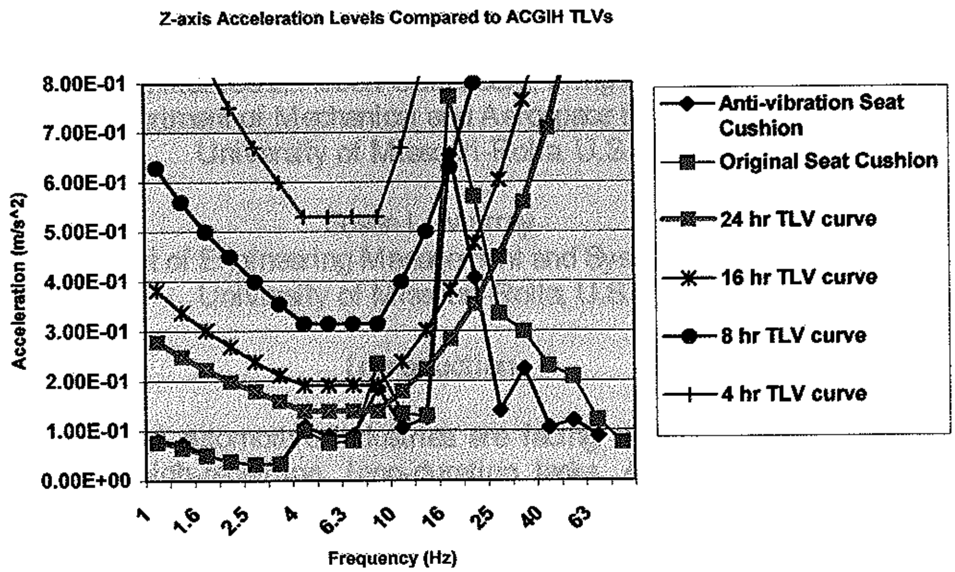
Current Hazard Reports indicated that pain in pilots' legs and backs begin two to four hours into the flight and increase with time. Mission readiness also decreases with an increase in flight duration due to the constant distraction of pilots shifting in their seats while trying to get comfortable. Froom, et al [2] reported a dose-response relationship between the length of military helicopter flights and back discomfort. He also concluded that this pain is typically dull, over the lower back, and its prevalence and intensity are dependent on the total flight hours of exposure.

Methods

This study evaluated WBV produced in the pilot seating systems onboard the MH-60S. The purpose of the study was to test and compare the effectiveness of two different seat cushions, the current seat cushion versus an anti-vibration seat cushion. Both seat cushions were measured for acceleration levels averaged over five-minute intervals using a triaxial seat pad accelerometer. The recordings were completed for a 3-hour straight and level flight. A frequency analysis from 0-80 hertz (Hz) was conducted on all acceleration measurements to determine the dominant axis and frequency of the pilots' vibration exposure. The results were then compared to the applicable Threshold Limit Values (TLVs) established by the American Conference of Governmental Industrial Hygienists (ACGIH) [1] and the International Organization for Standardization (ISO) 2631.1 [3] to determine the MH-60S pilots' permissible exposure time for both seat cushions.

Results

The results of the study showed that for both seat cushions the vibration levels of the z-axis at 16 Hz had the shortest allowable exposure duration, according to the ACGIH TLVs. In the z-axis at 16 Hz, the MH-60S's current seat cushion's acceleration levels indicated an exposure time limit of approximately 6 hours, while the anti-vibration seat cushion's acceleration levels pierced the 8-hour exposure time limit curve. This is shown in the graph below.



When compared to the ISO standard, the acceleration levels are 0.86 m/s^2 and 0.73 m/s^2 for the current and anti-vibration seat cushions, respectively.

Discussion

While the anti-vibration seat cushion's acceleration levels were slightly lower than the current seat cushion's levels, the helicopter pilots are still overexposed to WBV. Since the average flight during a deployment or mission could last up to 8 hours, the current exposure places the pilots at an unacceptable risk of injury, lack of mission readiness, and possible equipment damage. In the future, helicopters will be outfitted with auxiliary fuel tanks, enabling even longer flights.

Additional research should be conducted to include a larger sample size, evaluate specific flight profiles other than straight and level flights, and perform transmissibility studies aboard the MH-60S targeting specific portions of the human body. Additionally, extensive follow-up epidemiological studies should be performed for Navy helicopter pilots to evaluate the incidence rates of back injury and their relationship to whole body vibration exposure.

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